Power and The Engineer

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In writing upon a certain subject a correspondent says that "after a thorough discussion of the subject all parties concerned will have the same opinion as before." Is this a true statement of the frame of mind of the majority of engineers?

Is it true that instead of searching after the truth in the matter they seek rather to find reasons for believing that which may or may not be true, is true, because they wish it to be true?

It is true that thinking is hard work and it is also true that a new thought or a new idea almost invariably strikes the mind unfavorably. Steps from the beaten path of conventionality are taken reluctantly. But are engineers less progressive than other men?

Centuries ago someone said, "Can any good come out of Nazareth?" It was prejudice that asked the question that carried with it the implication that only an answer in the negative could be given.

Does prejudice dictate the position which engineers take on questions of policy or practice in their vocation?

If one is unwilling to put himself in another's place, to see as he sees, to get a view from the same position, to give full credit to ideas differing from his own, prejudice is the mainspring of his logic.

Are engineers whose daily contact with conditions and problems are always of such a practical nature, less able to take a practical view of any problem in their line than men in other walks in life?

In the discussion of any topic where full credit is not given to every scrap of evidence no conclusion can be reached which is entitled to confidence of any one, and least of all to the confidence of him who refuses to examine from a point of view other than his own.

It was prejudice that compelled Galileo to say that the sun revolved around the earth when he knew the reverse to be the truth.

It has many times been shown that blind unreasoning prejudice has clogged the wheels of progress and embittered the lives of some of the world's greatest men. It is born of ignorance and lives only until the light of investigation brings knowledge. As the ripple spreads from shore to shore and lasts long after the stone that made it has gone to the bottom, so is the influence of prejudice felt long after its cause has disappeared. Prejudice, favorable or unfavorable, is to be avoided like the pestilence, for it always leads to the wrong road and to the retracing of numberless wasted steps.

Engineers, above all men, should be free from this character-destroying influence of prejudice.

"Prove all things."

Central Station at Newcastle, Indiana

By Osborn Monnett

In this small central power station of only 700 kilowatts capacity, the exhaust steam is used to supply a hot-water heating system to neighboring customers. The main features of the plant are its simplicity and its adaptability to the existing requirements.

It is not always necessary to go to the large metropolitan power plants in order to find interesting things in steam engineering; frequently the smaller plants show the result of just as much careful forethought as any of the large and more prominent installations. Whether in a large or small plant the best solution of any engineering problem is that which will realize the largest returns upon the investment under the existing conditions.

The plant of the Newcastle Light, Heat and Power Company is a small central station containing compound-noncondensing engines; for during the greater part of the year the exhaust steam can be more profitably utilized for heating than by exhausting into a condenser. The location of the plant is such as to simplify the coal-handling problem, it being placed adjacent to the elevated tracks of the Pennsylvania railroad, a spur of which runs directly over the coal bunkers at an approximate hight of 16 feet above the boiler-room floor. Fig. 1 is an exterior view of the power house, showing a carload of coal on the elevated track. From the bottoms of the dump cars the coal falls into a storage pocket of 400 tons capacity, from which point it descends, partly by gravity, into hand cars at the boiler-room floor. These cars each have a capacity of 1800 pounds of coal and every load is weighed on a platform scale before being delivered to the boilers. Fig. 2 is a view of the boiler room, showing the hand cars standing against the coal-pocket bulkhead.

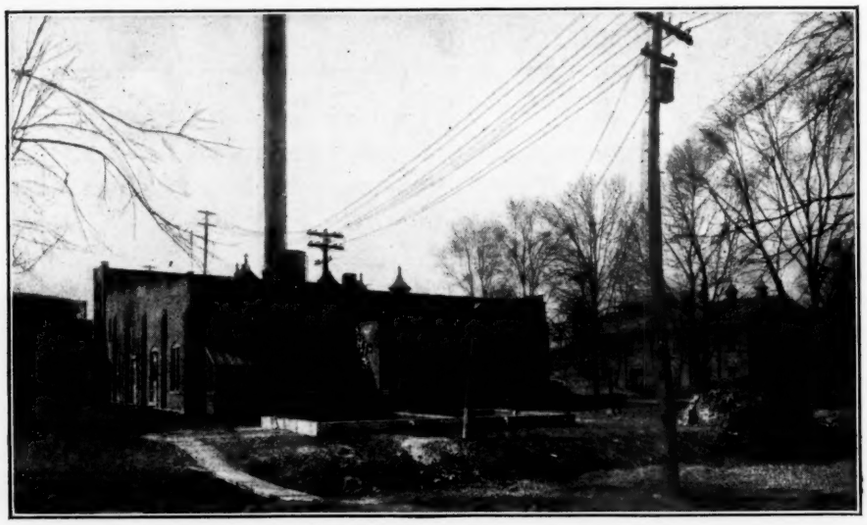


FIG. 1. EXTERIOR VIEW OF PLANT

The plant, as shown in the sectional elevation, Fig. 3, consists of four separate compartments, the coal pocket, the boiler room, the pump room and the engine room, arranged in the order named. Four 250-horsepower hand-fired Stirling boilers are installed, discharging the products of combustion into an underground flue and thence to a steel stack 6 feet in diameter and 150 feet high, lined to the roof of the boiler room with firebrick. Two of the boilers are equipped for burning natural gas, if desired, in addition to using coal for fuel.

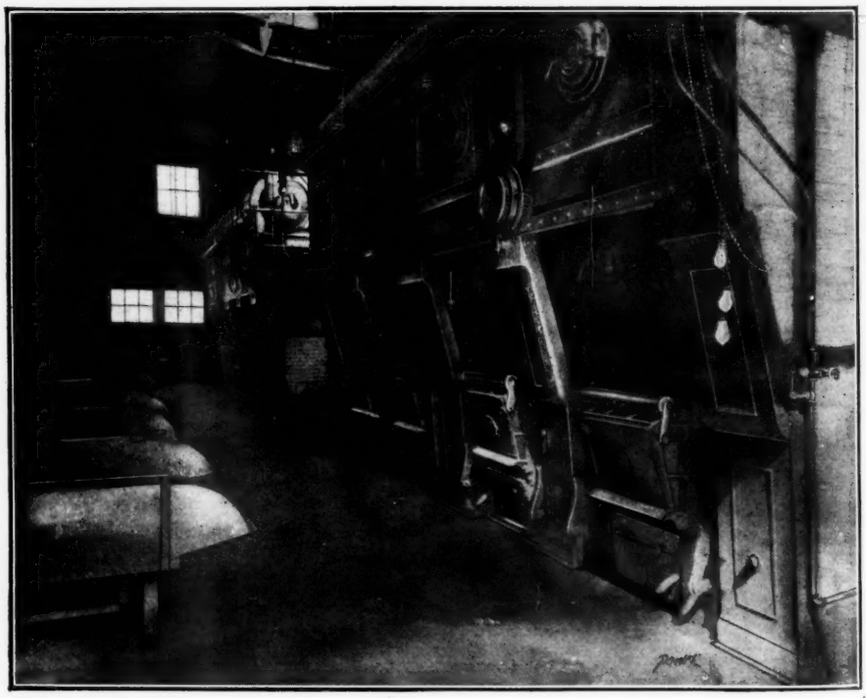


FIG. 2. BOILER ROOM

Six-inch long radius bends lead from the boilers to a 10-inch header situated in the pump room, from which branches extend to the engines. The header contains three drip pockets for catching condensation, and in addition there is a separator on each engine line, the condensed steam being taken care of by Morehead tilting traps and returned to the heater.

In the engine room there are three generating units: two 200-kilowatt cross-compound Russell engines, each direct connected to a two-phase, 60-cycle, 2300-volt Western Electric generator, to which is belted an exciter; and a third Russell engine and generator of 300 kilowatts capacity. Each engine has an independent gravity oiling system which is shown in the view of the engine room, Fig. 4. The electrical load consists of both lighting and power circuits, the distributing circuits being at 2300 volts, which is stepped down to 220 volts for factory lighting and power and to 110 volts for the lighting of private residences. A satisfactory power load has been developed which, with the night lighting load, enables the plant to run continuously 24 hours a day.

One of the interesting features of the electrical equipment is the location of the switchboard. This consists of blue Vermont marble panels carrying the usual switches and recording instruments, but the panels are set flush with the engine-room wall so that the board takes up no room in the engine room proper, although

it is accessible from the back, which opens into the electrical-repair department and meter-testing room, where all of the switchboard connections are easily available for inspection or repairs.

A point in which this plant differs from the ordinary small central station is in the hot-water heating system by which practically all of the exhaust steam is utilized for profit. Therefore, the pump room in which this part of the apparatus is located is of considerable interest. A view of the pump room is shown in Fig. 5. The question of water supply is one of importance in this class of plant as the system must be kept fully charged at all times to enable the full benefit to be realized from the heating mains. Three driven wells, 4 inches in diameter and 43 feet deep, furnish all the necessary water. The water in the wells usually stands at a level of about 7 feet below the ground and by means of an air lift it is raised and discharged into a concrete reservoir of 18,000 gallons capacity. This reservoir, which is shown in the foreground in Fig. 1, is built in two compartments and stands a few feet above the level of a creek which runs through the property; hence, either compartment may be drained by gravity and cleaned while the other compartment is in use. This makes it possible to use chemicals for treating the water before it enters the system, if such a practice is found desirable.

From this reservoir the water is pumped to a 4000-gallon steel tank situated on the roof of the power house and giving a head of about 35 feet, from which the water is fed automatically to a 500-horsepower Hoppes feed-water heater located in the pump room. The boilers are fed by a 10 and 6 by 12-inch Dean pump, and two 7½ and 5 by 6-inch Blake pumps. Two separate and distinct feed lines are arranged, enabling one to be entirely taken down for repairs while the other is being used.

The exhaust steam from the engines and also from all pumps is collected in one large exhaust main and passes through either one of two Baragwanath heaters through which is circulated the water for the heating system. The water leaves the power house through an 8-inch main, provided with expansion joints, wrapped with mineral wool and inclosed in wooden conduits about three feet below the surface. The pressure is kept up on the system by two 12 and 10 by 18-inch Stillwell-Bierce pumps, the outgoing pressure being about 60 pounds and the incoming pressure 30 pounds. Under ordinary conditions the water loses from 20 to 30 degrees in making the circuit.

An important part of the system is the make up pump and its connections, which may take water from the open heater, the house tank or the reservoir, and discharge it into the suction of the circulating pumps. This pump, as well as all other pumps in the power house, is controlled by a Carr pump governor and works automatically.

In addition to the heating equipment described, provision is made so that one of the boilers may be connected in series in the hot-water line by which means can be supplied any necessary additional heat in excess of that supplied by the exhaust steam. It has been found that in ordinary weather, practically all of the exhaust steam is condensed in the heaters, thereby giving up all of its energy for useful work. The exhaust line is well protected by oil extractors so that the condensed steam may be used for feed water.

The plant was installed by the Arbuckle-Ryan Company, of Toledo, and by its simple design it is well adapted to the particular service, whereas a more complicated and costly equipment would not have solved the problem as well.

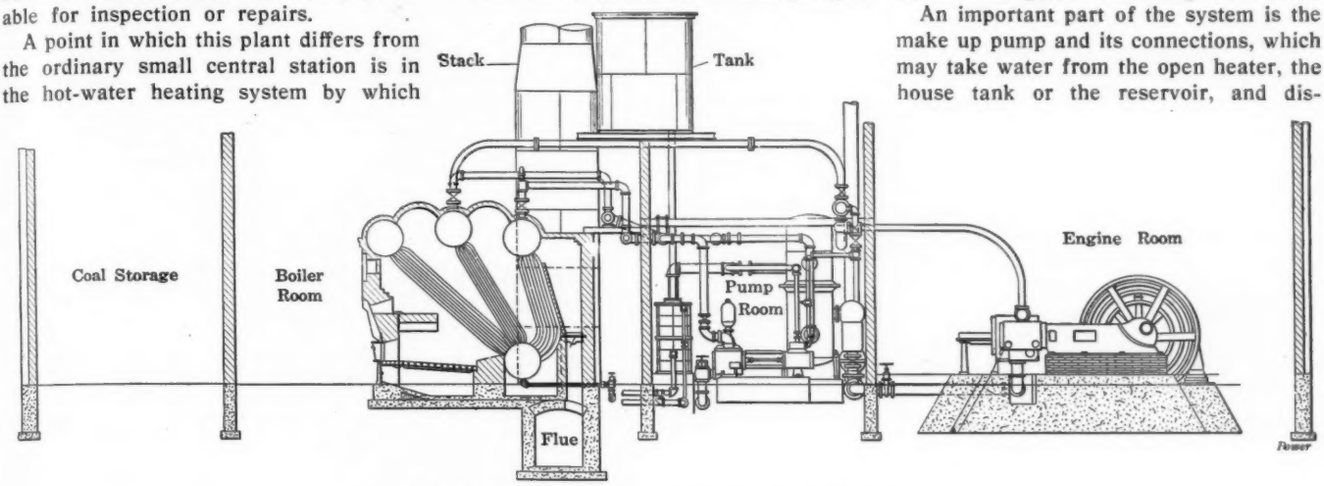


FIG. 3. SECTIONAL ELEVATION THROUGH PLANT

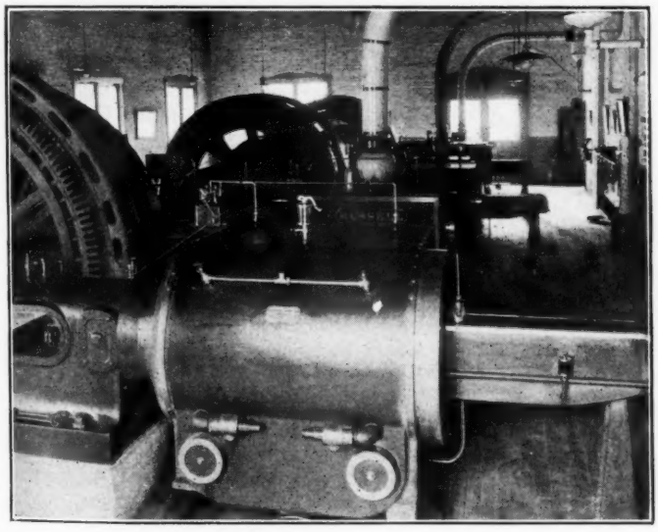


FIG. 4. ENGINE ROOM

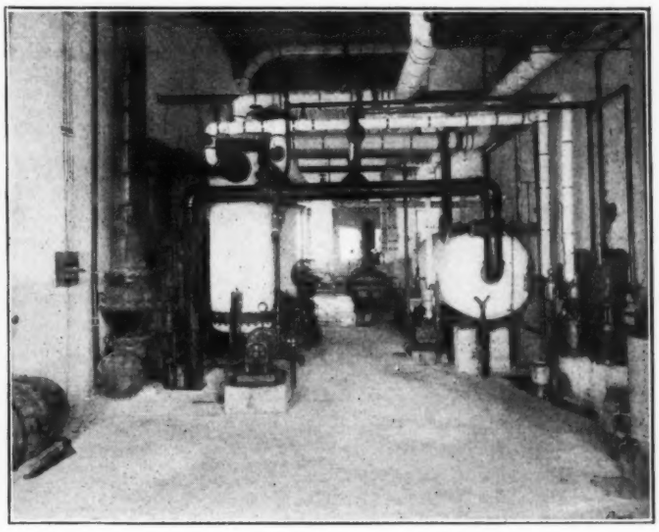


FIG. 5. PUMP ROOM

The Steam Turbine in Germany

By F. E. Junge and E. Heinrich

Increase in use of steam and water power in Prussia, Switzerland and the United States. Increase in economy of the steam engine and factors influencing development. Three principal types of axial turbines and the tendency to merge into one standard design embracing the desirable features of each.

THE EVOLUTION OF POWER GENERATION

The per capita production of power in a certain country affords a fair measure for the condition of power generation, technical progress and for the industrial ability of the people inhabiting it. The diagram in Fig. 1, which, like the others in this article, is taken from a paper communicated to the Society of German Engineers by Professor Kammerer, Charlottenburg, shows that the curve of steam-power generation in Prussia is almost a straight line, rising from 0.055 horsepower per capita in 1890 to about 0.14 horsepower in 1907. (In the following, metric horsepower is used exclusively. It should be remembered that one metric horsepower equals 0.986 English horsepower.) The steam-power curves for the United States and Switzerland, which are given for comparison, show similar tendencies, while the development of water power is much more marked in Switzerland than in the States, for the simple reason that the cost of fuel in the latter country is comparatively low, while the former is absolutely dependent on her water-power development for industrial production.

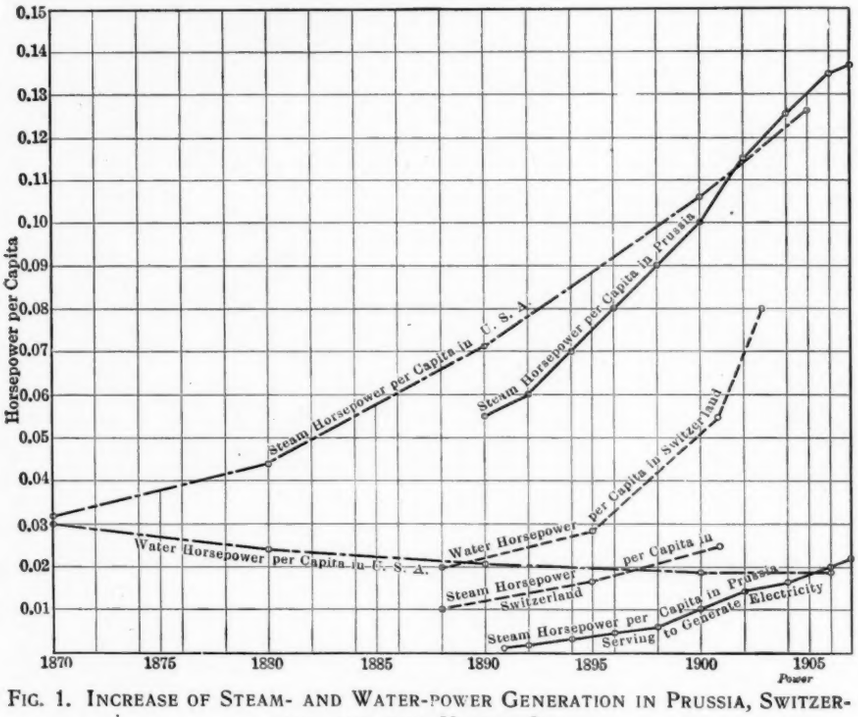


FIG. 1. INCREASE OF STEAM- AND WATER-POWER GENERATION IN PRUSSIA, SWITZERLAND AND THE UNITED STATES

The lowest curve in the diagram shows what portion of steam power in Prussia is transformed into electric energy. This curve, too, is in the ascent, revealing the growing tendency in that country to effect the distribution of power by means of electric energy instead of by individual generating plants. As a matter of fact, we are coming more and more to the point where power will be generated at the mouth of pit or other convenient places, using inferior grades of coal unfit for transportation, whence it is distributed in a high-tension net covering and supplying the whole country. The figures relating to power production in Prussia are taken from the latest official statistics, but they refer to the condition of plant when the latter is for the first time put into practical operation. As the load factor increases with the growing demand for power, the plant output rises in proportion. Hence, the capacity of power production is in reality considerably higher than the curve would imply.

Technical and commercial progress is illustrated further by the improvements in the *efficiency* of energy transformation. When making comparisons, however, we must embrace in our conclusions a consideration of the differences in the geographical structure of the different countries, which will make the attainment of highest economic excellence less desirable, for instance, in industrial circles in the United States, as long as coal, oil and natural gas are available in sufficiency, than in the industries in Germany, where natural limitations and a dense population force the engineer to strive for the utmost utilization of meager resources. All points considered, we may say, briefly, that the success of the German industry on the world's markets rests on the principle: get quality, rather than on the demand: get tonnage.

The diagram in Fig. 2 shows the actual consumption of heat in steam engines from the year 1860 up to 1908. In the first-named year the consumption per horsepower-hour was over 11,000 calories or 43,560 B.t.u. (single-cylinder engine working at 2.5 atmospheres or 36.7 pounds per square inch boiler pressure). By raising the steam pressure to 5.5 atmospheres the heat consumption of a single-cylinder engine was reduced to 7600 calories or 30,096 B.t.u. per horsepower per hour in 1874. The marked reduction of heat consumption to 6400 calories or 25,344 B.t.u. was effected in the same year by the introduction of the compound engine. In the following years the heat consumption per horsepower-hour decreased gradually to 5000 calories or 19,800 B.t.u., while the boiler pressure at the same time increased to 8 atmospheres or 116.8 pounds per square inch.

A further reduction was effected by the advent of the triple-expansion engine in 1890, with a corresponding rise in steam pressure to 11 atmospheres or 161.7 pounds per square inch. During all of that time the temperature of the steam had been hardly changed because saturated steam was used almost exclusively up to 1890. It was only on account of the difficulties experienced when trying to raise the steam pressure beyond 13 atmospheres or 191 pounds that the superheating of the steam was first attempted. The result was a further reduction of heat consumption to 3300 calories or 13,068 B.t.u. per horsepower per hour, attained with a triple-expansion engine of 2000 horsepower in Görlitz. This, it seems, represents now the upper limit of steam temperatures for practical operations. Attempts to go still higher have failed on account of the difficulty of constructing durable superheaters.

At this point, when the development of the reciprocating engine seemed to come to a halt, steam-power generation received a new impetus by the development of the steam turbine. While the earlier types of this prime mover showed a higher heat consumption than the best steam engines, on account of the low

temperatures and pressures originally employed, it has been possible of late to bring the thermal efficiency of steam turbines to the same level of excellence attained by their competitor. Regarding heat consumption, both types are now practically on a par. But the steam turbine has brought about economies in a new direction: lowering the cost of buildings, floor space, attendance and lubrication.

The diagram in Fig. 2 contains, besides the curves showing actual steam consumption, boiler pressure and steam temperature, a curve giving the theoretical heat consumption of an ideal engine, working with the same initial temperatures as the rest. The ordinates between these two curves represent the loss accruing through imperfections in the engine, said loss having been reduced from 9000 calories or 35,640 B.t.u. in 1860 to 2000 calories or 7920 B.t.u. in 1908. The straight line running parallel to the abcissa axis of the diagram represents the heat consumption of an ideal engine working with an initial temperature of 1000 degrees Centigrade. The ordinate fragments between this line and the curve of theoretical heat consumption represent the loss due to the employment of low temperatures. Thus, the diagram in Fig. 2 gives a graphic history of the development of steam-power generation as far as heat consumption is concerned. It shows that inventive talent will seek and find new outlets if natural obstacles hamper progress in the direction originally pursued. It shows further that the art of steam-power generation is still far from perfect and that its practioners will have to concentrate their best efforts in order to compete successfully with other factors such as gas, oil and water power, none of which has reached its highest level of excellence.

The diagram in Fig. 4 shows in a similar way the development of the reciprocating steam engine for ship propulsion from 1850 to 1890, according to tests made on the steamers of the two largest German shipping companies: The Hamburg-American Line and The North German Lloyd.

The influence of the size of engine on the cost of steam power is shown in the diagram in Fig. 3. Experience has taught the designer that large engines can stand a careful working out of details much better than small engines, without considerably increasing first cost. Thus the upper limit of all-round economy is greatly raised by simply enlarging the capacity of heat engines. The diagram in Fig. 5 shows the effect of prime-mover capacity on the factor of heat consumption. The latter decreases and economy increases with growth in the size of the plant. Some of the later thermotechnical improvements, for instance superheating in steam engines and increasing compression pressures in gas engines, have benefited small and large engines alike. But large-scale power generation has the great advantages of lower initial cost and lower cost of attendance as well as cheaper supply of fuel.

The operating cost of heat engines was reduced among other things by the adoption of automatic lubrication. With steam turbines, forced lubrication was a necessity in order to supply oil to the bearings under pressure. But, in reciprocating engines, ring and central lubrication have been adopted with an equal advantage. The cost of attendance was further greatly reduced by the introduction of automatic stoking. In all improvements which regard the reduction of power cost by the elimination of the personal element or, generally speaking, in the replacement of human labor by mechanical means, American industry has been our teacher, while in all cases which regard the raising of economy by the rational utilization of matter, of fuel and other raw materials the German industry has been in the lead. In the one case scarcity and high cost of skilled labor, in the other scarcity and governmental control of natural resources were responsible for the results attained, an example which seems to prove that technical progress does not travel along the path of least resistance, but attains highest excellence when it is compelled to overcome obstacles.

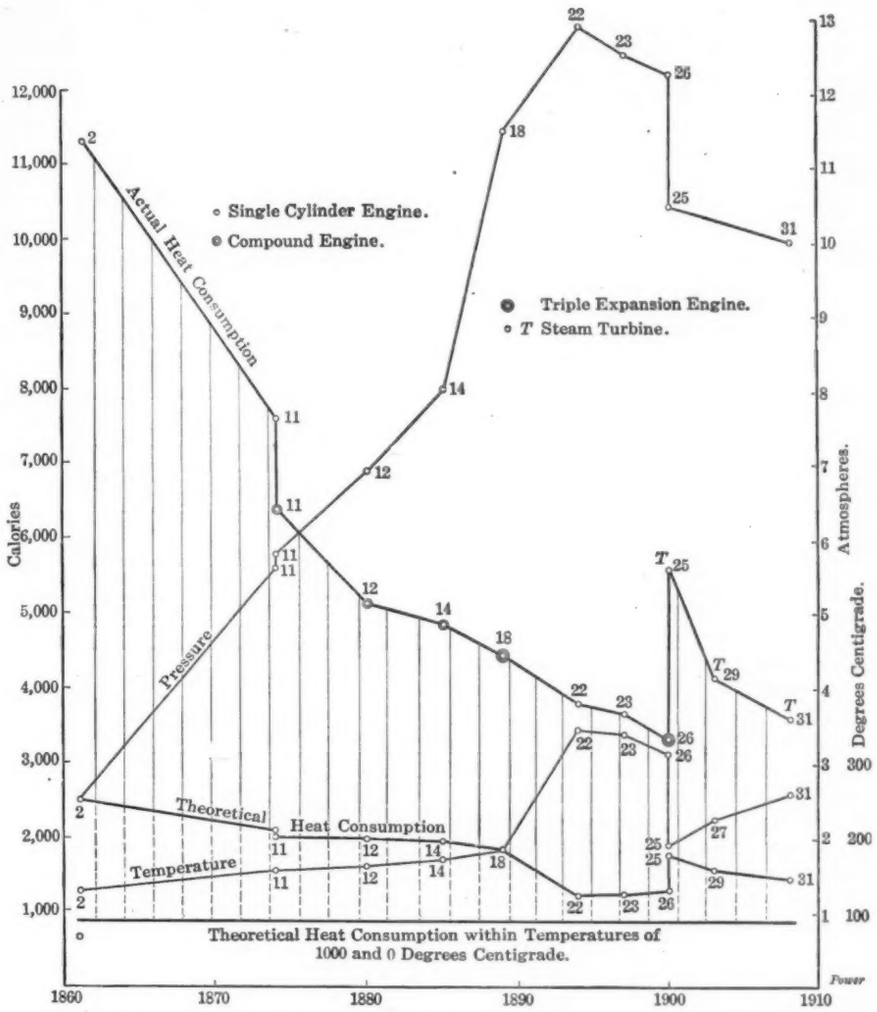


FIG. 2. HEAT CONSUMPTION OF STEAM ENGINES AND TURBINES FROM 1860 TO 1908

No. 2. Single-cylinder marine engine from 1861, 204 h.p., 17 r.p.m.
No. 4. Single-cylinder marine engine from 1874, 282 h.p., 62 r.p.m.
No. 11. Compound marine engine from 1874, 267 h.p., 71 r.p.m.
No. 12. Compound stationary engine from 1880, 132 h.p., 71 r.p.m.
No. 14. Compound balance pumps from 1883, 251 h.p., 13 r.p.m.
No. 18. Triple-expansion engine from 1889, 200 h.p.
No. 22. Triple-expansion engine from 1895, 1042 h.p., 60 r.p.m.
No. 23. Compound Schmidt engine from 1899, 100 h.p., 81 r.p.m.
No. 25. Steam turbine from 1900, 1000 kw., 1500 r.p.m.
No. 26. Triple-expansion engine from 1900, 2000 h.p., 83 r.p.m.
No. 29. Steam turbine from 1903, 1000 kw., 1500 r.p.m.
No. 31. Steam turbine from 1908, 3500 kw., 1360 r.p.m.

The diagram in Fig. 6 shows the economic result in a boiler plant of the adoption of coal conveyers and automatic stoking. Formerly there were required in this plant 54 firemen and 2 overseers, necessitating an outlay in wages of 0.164 mark or 3.9 cents per ton of steam. After the introduction of mechanical means only 20 firemen, 2 overseers and

2 machinists were required. The number of high-class workmen was doubled, while the number of unskilled laborers was reduced in the proportion of 2.5 to 1.

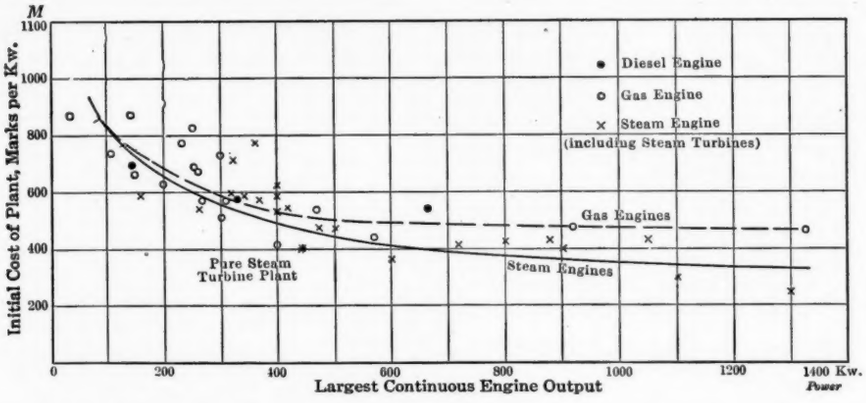


FIG. 3. RELATION OF INITIAL COST OF STEAM ENGINES AND TURBINES TO SIZE OF PLANT

The amount of wages paid decreased to 0.061 mark or 1.45 cents per ton of steam. To this must be added the cost of upkeep, interest and amortization for chain grates and conveyer bands amounting to 0.034 mark, so that the total cost was reduced to 0.095 mark or 2.3 cents, which is almost two-thirds of the original amount. This saving was effected by employing, instead of unskilled labor, automatic machinery and high-class labor. Of course, owing to the somewhat cheaper and more reliable labor which we have in Germany, the replacing by mechanical means of the human element in industrial plants is not always so profitable as it is in the States. But the general tendency of power generation is the same in our country as it is in America: concentration of production in large plants in order to secure utmost commercial economy.

DEVELOPMENT OF THE STEAM TURBINE

In Germany the steam turbine had to compete with a highly developed steam engine as well as with the gas engine, whereby its coming has been somewhat delayed, and even today it has to meet the growing competition, especially for ship propulsion, of a dangerous rival: the Diesel oil engine. Not until 1898 was the first large steam turbine, a 1000-kilowatt Parsons turbine, erected in the electric central station of Elberfeld, where it has given complete satisfaction. The following rapid evolution is notable for the desire on the part of manufacturers to attain highest possible fuel economy rather than simplicity of construction and operation. The result of this endeavor is best illustrated by the fact that a consumption of 5.4 kilograms or 11.9 pounds of steam per kilowatt-hour is attained with a 4000-kilowatt steam turbine installed by the Allgemeine Elektrizitäts Gesellschaft in the central station at Rummelsburg, Germany.

It is an acknowledged fact that the development of steam-turbine construction, after having acquired a high degree of thermal excellence, tends now toward a standardization of the different systems, which at present are dominated and characterized by a few simple principles of steam flow and structure. The following discussion of these principles will be confined merely to axial turbines, because the radial turbine has not so far shown an equal capability of development, at least for purposes of large-scale power production.

First. The many-stage reaction principle developed by C. A. Parsons. As regards the utilization of steam energy this system is characterized by the fact that the conversion of the potential energy of the steam into kinetic energy takes place not only in the guide-blade channels, but also in those of the rotating part. Regarding construction, the Parsons system is characterized by the use of a drum with dummy pistons and labyrinth packing.

Second. The many-stage action principle with one row of blades on each rotating wheel, developed by Rateau and modified by Zoelly. As regards the utilization of steam energy this system is characterized by the fact that the conversion of the potential steam energy into kinetic energy takes place in the guiding apparatus only, while the whole kinetic energy of a stage is utilized in a single row of rotating blades. Regarding construction, the Rateau-Zoelly system is characterized by the employment of several disk wheels and by the separation of the different stages through guide-wheel disks, which reach unto the hubs of the rotating wheels.

Third. The many-stage action principle with velocity staging developed by Curtis. As regards the utilization of

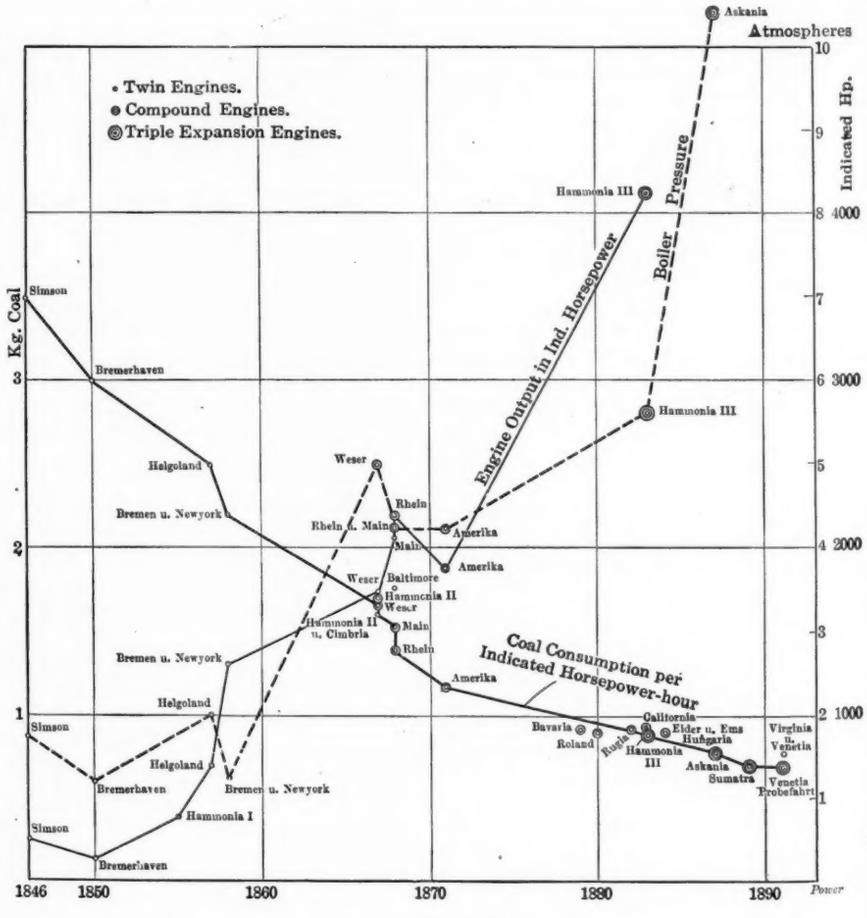


FIG. 4. COAL CONSUMPTION OF NORTH GERMAN LLOYD AND HAMBURG-AMERICAN LINE STEAMERS FROM 1850 TO 1890

steam energy, this system is characterized by the fact that the whole potential energy of one stage is, as a rule, transposed into kinetic energy in expanding nozzles, and further, that the kinetic energy of each stage is utilized in several rows of revolving blades. Regarding construction, the Curtis system possesses rotating disk wheels, like No. 2, but fewer in number, which are equipped with several rows of blades; in addition, it possesses deviating blades which are fixed in the casing. (Some designers do not provide for the complete expansion of the steam in the nozzle, but transpose part of the potential energy in the first row of revolving blades in order to attain higher efficiency. But this feature is not generally practised and the amount of steam converted is quite small, so that the above classification is approximately correct.)

The principle of one-stage action has not been included in the above. This system, as developed by De Laval, shows in its practical execution several remarkable solutions; for instance, the flexible shaft and the disk wheel with high peripheral velocity. But the very nature of the De Laval turbine, at least in its original form, militates against the attainment of higher capacities. The high peripheral velocity and the resulting high wheel diameters and best material. But practice has evinced that buckets milled into the rim of the wheel would not stand the eroding action of the steam jet, and that the steam consumption increased very materially owing to the destruction of the sharp admission edges of the buckets.

The three above-mentioned fundamental principles of steam-turbine construction have developed each for itself in Germany. The representative firms of each are: First, of the Parsons system: Brown, Boveri & Co., Baden, Switzerland, and Mannheim, Germany, licensee of C. A. Parsons; second, of the many-stage action system: the Zoelly syndicate, embracing six of the leading firms in Germany; third, of the Curtis system: Allgemeine Elektrizitäts Gesellschaft, (A. E. G.), Berlin. Gradually there has been a fusion of the different systems. For example, the A. E. G. has abandoned the Curtis construction of the low-pressure part of its 1500 revolutions per minute type and has adopted the principle of many-stage action as developed by Rateau and Zoelly. Up to this year the pure Curtis principle was followed in its 3000 revolutions per minute type, showing two pressure stages with two velocity stages each. The best known tains only two rotating wheels with two rows of buckets each and only one guide-wheel disk, is lower than the cost of a turbine of equal output which contains, besides a rotating wheel with two rows of buckets, from five to seven wheels with one row of buckets and the same number of guide-wheel disks. But the severe demand for a low steam-consumption guarantee, which would enable the firm successfully to compete with other German makes, forced the designer to adopt a somewhat more expensive construction. The Bergmann Electric Works build their turbine similarly, using a combination of the Curtis and Rateau principles.

Even the foremost of the German firms representing the Parsons interests, Brown Boveri & Co., have, after long hesitation, resolved to follow the example of other builders of reaction turbines. They have replaced the high-pressure stages of the Parsons turbine by a Curtis wheel, reducing the length and weight of the unit enormously. Only for large units and special-service conditions is the pure Parsons principle maintained. Last, but not least, the Curtis principle begins to intrude into the Zoelly syndicate, which, so far, has resisted with tenacity all attempts to change its original mode

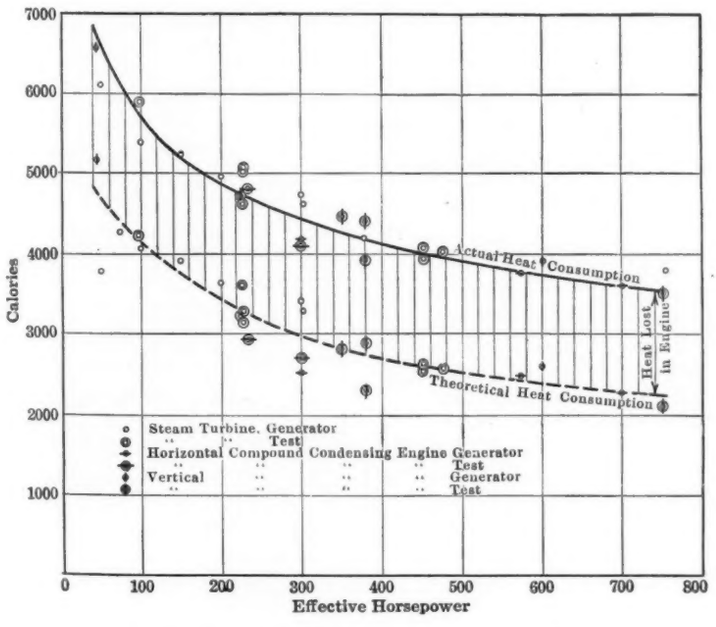


FIG. 5. INFLUENCE OF SIZE OF ENGINE ON HEAT CONSUMPTION

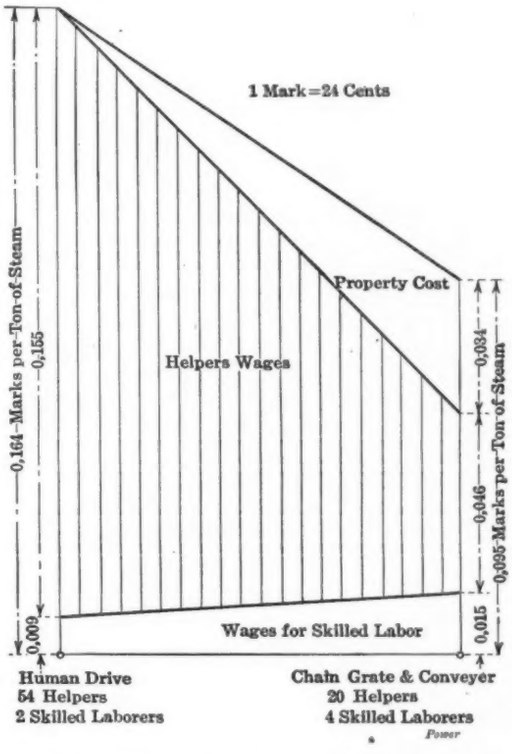


FIG. 6. REDUCTION OF STEAM COST IN BOILER HOUSE BY ELIMINATION OF HUMAN LABOR

number of revolutions necessitate the employment of gears in order to get down to practicable speeds. Gears for high output have not, however, found favor in stationary practice. In the Riedler-Stumpf turbine the attempt was made to attain reasonable speeds by adopting very large type of this class is the turbine of 1000-kilowatt capacity. But for outputs beyond 1000 kilowatts the A. E. G. now builds the low-pressure part after the Rateau-Zoelly principle. Evidently the cost of construction of the said type of 3000 revolutions per minute, which con- of construction. The Augsburg Nürnberg Works, members of the syndicate, have just begun to build all of their 3000 revolutions per minute types on a combination principle, using the Curtis and Zoelly features. The Görlitz Machine Works, licensees of Zoelly, have also

built normal types of turbines in accordance with the above principle, but only for sizes below 700 kilowatts. The other members and licensees of the Zoelly syndicate have not adopted the new practice.

Summing up, it may be said that turbine construction tends toward the creation of a standard system of unity, which will include all of the best features of the different types. In the present stage of development the following variations exist: First, the combined Curtis and Rateau system; second, the combined Curtis and Parsons system; third, the pure Rateau-Zoelly system; fourth, the pure Parsons system; fifth, the pure Curtis system. The above order of importance has been arranged according to the influence which, in our notion, the various systems exercise upon the art of turbine building for stationary purposes in Germany today. Which of the systems will finally come to the front the competition of the future will evince.

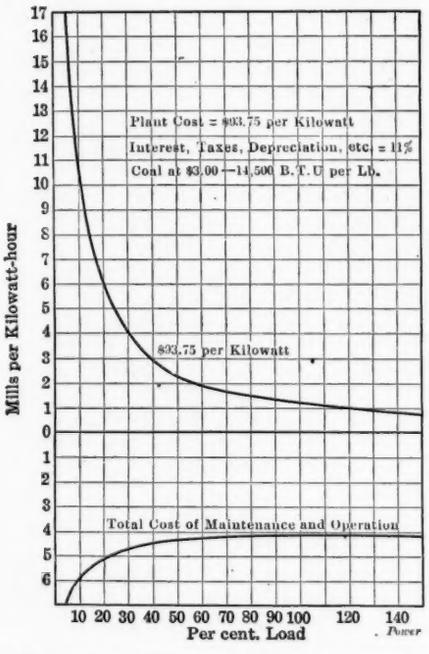
Carrying Peak Loads Economically*

By H. G. Stott

The solution of this problem is to reduce the investment per kilowatt to the minimum. This may best be effected at present by the use of steam turbines, large grate area and forced draft.

In any electric railway the first requisite of the system is that the power supply shall be absolutely reliable, and the second is that it shall be produced economically. The first principle is so thoroughly understood that nothing further need be said in regard to it. The second, however, involving *all* the cost of power, is not so clearly understood as might be wished, especially as regards peak loads.

The curves shown in the accompanying diagram show the total maintenance and operating costs per kilowatt-hour for various percentages of load factor *below* the horizontal axis, while the curve *above* this axis shows the fixed charges per kilowatt-hour for various load factors. The sum of the ordinates above and below the axis give the total cost of power under the conditions which have been assumed.


Plant Cost = $93.75 per Kilowatt
Interest, Taxes, Depreciation, etc. = 11%
Coal at $3.00—14,500 B.T.U. per Lb.
$93.75 per Kilowatt
Total Cost of Maintenance and Operation
Mills per Kilowatt-hour
Per cent. Load
Power

OPERATING COSTS AND FIXED CHARGES FOR VARIOUS LOAD FACTORS

From a consideration of these curves it will be seen that for peak loads having an annual load factor of from 5 to 15 per cent., the operating and maintenance charges are relatively unimportant and the fixed or capital charges are of the greatest importance.

The various sources of power to be considered for peak-load purposes are as follows:

1—Storage batteries.
2—Purchased power.
3—Hydroelectric power.
4—Gas engines.
5—Steam turbines.

1. Storage batteries would seem at first sight to be an ideal means of carrying peak loads, as obviously it would not call for any additional plant capacity, but if we apply our double curves to it, we will find that the fixed charges, on a battery capable of discharging at its maximum rate for two hours at a time, are so high as to leave it out of the question.

2. Purchased power almost invariably carries a heavy fixed charge per kilowatt of maximum demand, varying from $15 to $25 per kilowatt per annum, with a further charge per kilowatt-hour actually used. The first charge should be treated as a fixed charge and so plotted above the axis. This will in all probability show an extremely high cost per kilowatt-hour of peak load.

3. Hydroelectric power, if transmitted more than a few miles, will inevitably show very high fixed charges, owing to the large investment in the power plant, hydraulic development, transmission lines, etc.

4. Gas engines, while attractive from the point of view of small cost for fuel during the period the plant is idle, are inadmissible from the point of view of high fixed charges due to the large investment per kilowatt.

5. Steam turbines are at present the most satisfactory type of prime mover in the market for the peak-load problem, owing to their low first cost. If a reciprocating-engine plant in good condition is available, then the addition of low-pressure turbines will result in the development of a kilowatt at an investment of about $25, provided there is enough room available in the engine room and if the boilers can be forced sufficiently to carry the extra load.

In the boiler room the investment can be kept down by adding grate surface instead of more boilers, and by the use of forced draft the old rating of 10 square feet of heating surface per boiler horsepower can safely be reduced to 4 or 5 without materially adding to the cost of boiler or furnace maintenance.

While the overall boiler efficiency will begin to fall off gradually beyond 175 per cent. or 200 per cent. rating, the small loss thus entailed is insignificant compared to the saving in fixed charges as shown in the diagram.

The solution of the problem of carrying peak loads economically is therefore to be found in reducing the investment per kilowatt to a minimum and this can be best accomplished at present by the use of steam turbines and by the use of large grate area, such as a ratio of 30 or 40 square feet of heating surface to each foot of grate area, instead of the present ratio of 55 or 60 to 1. Forced draft may be employed with advantage with any grate area and will in almost any case result in increased economy as well as capacity.

*Portion of the report of the Committee on Power Generation of the American Street and Interurban Railway Engineering Association, Atlantic City, October 10-14.

A travelin' feller kum in t' Bill Grimes' ingin room tother day an' told him th' old yarn 'bout th' feller thet sed he'd diskivered perpetshul motion an' thet it wuz his mother-in-law's tongue. Bill thot it wuz sich a good joke that he sprung it on his mother-in-law.

Th' docter thinks they kin patch Bill up so he kin get out uv th' hosspittle in 'bout three weeks.

New York City is still the leader by a long margin in the number of licensed engineers, there being nearly 13,000 It is doubtful if there are as many stationary engineers in any other State, excepting Massachusetts, Ohio and Pennsylvania.

Boiler Tube Soot Cleaners

By Warren O. Rogers

For years tube scrapers have been used to remove soot from boiler tubes. Many kinds, good, bad and indifferent, have been designed, and of these twenty-seven types are briefly described in this article.

The formation of clinkers on a grate not only makes the fireman's work harder, but prevents proper combustion from taking place, as the amount of air passing up through the grate is limited. Imperfect combustion causes smoke, which deposits soot on the boiler tubes, and this reduces the efficiency of the heating surface of the boiler.

A fireman can immediately notice the effect of removing the clinkers from the

FIG. 1

grates, because the fuel burns more freely, the steam pressure is more easily maintained and less attention is required in handling the fires.

An accumulation of soot on the heating surface of a boiler should be avoided just as much as clinkers on the grates. The fireman may not think much about its influence in preventing heat transmission through the tubes to the water, but if the sooty tubes of a boiler are cleaned during running hours the result will be at once manifest.

Just to what extent soot prevents heat transmission through the metal is not definitely known. This is a matter that seems to have been neglected, but even so it is no reason why the individual fireman should not keep his boiler or boilers as free from soot as the running conditions will permit.

The result of boiler tests recently conducted showed that with soot on the tubes of a boiler the evaporation from and at 212 degrees per pound of coal was 6.2 pounds of water, 13.4 pounds of coal being burned per square foot of grate surface per hour. The temperature of the gases escaping up the stack was 627 degrees Fahrenheit.

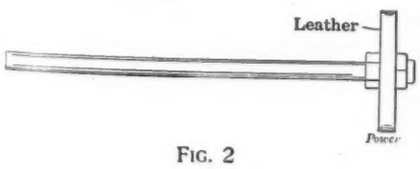


FIG. 2

With the tubes cleaned, an evaporation from and at 212 degrees per pound of coal of 7.04 pounds of water was obtained, which is a gain of 13.5 per cent. over the soot-coated tubes. The coal consumed per square foot of grate surface per hour was reduced from 13.4 to 9.09 pounds and the chimney gases reduced to a temperature of 546 degrees, or lowered 81 degrees.

Boiler tubes should be cleaned at least once in every twelve hours' run, if soft coal is used. Various methods are used for cleaning tubes. The most simple devices are the tube brush and tube scraper. Steam soot-blowing and sucking devices are also extensively used and lately several types of permanently fixed soot blowers have been devised. Many of these types of soot-handling devices will

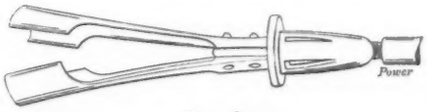


FIG. 3

be described and illustrated in this and following articles.

Probably the first boiler attendant who recognized that the transmission of heat through boiler tubes was retarded on account of soot, and decided to use a tube cleaner, tied a bunch of rags to a length of piping or a stick, something as shown in Fig. 1, and, by pushing the bunch of

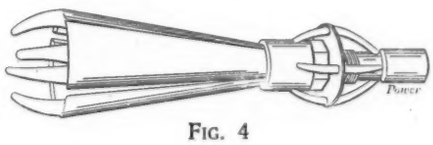


FIG. 4

rags through the tubes, forced some of the soot out into the combustion chamber.

Better results from cleaner tubes led to a slight improvement in design, such as Fig. 2, which might have consisted of a leather washer secured on a rod by means of two nuts. The leather washer, being stiff, would give better results than the rag arrangement. But as soot readily adheres to the inner surface of the tubes, these crude devices proved inadequate, and, as inventors realized the necessity of producing an efficient tube scraper, many designs have been brought out.

Fire-tube-boiler soot cleaners may be grouped into two general classes, soot tube scrapers and soot steam blowers and suckers.

An early type of scraper is shown in Fig. 3. It is self-explanatory. The scraper knives cut the soot, and the disk at the inner end of the scraper blades pushes the loosened soot out of the rear end of the tube. As the blades required springing together by the hand of the operator before entering a tube, such a scraper could not be expected to receive a welcome in the fire room.

The scraper, shown in Fig. 4, is made by the Gem Manufacturing Company. It consists of four cutting blades, each of which is fitted with a curved finger so

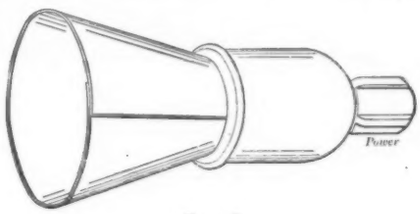


FIG. 5

that the cutting knives will readily enter the tube. The blades are made of tempered steel. When the scraper is inserted in a tube the blades get under the soot scale and remove it from the tube surface. The disk at the base serves two purposes, that of drawing out the loose soot on the return passage of the scraper, and of forming a support for the curved ribs which keep the scraper head in line when being pushed through a tube.

Fig. 5 illustrates a scraper made by the Gilmore Tube Scraper Works. The cutting edge consists of a steel cone. A sleeve moving on a threaded shank contracts or expands the cutting edge of the cone, and a socket to carry the handle acts as a lock nut to hold the sleeve in place. The elasticity of the steel cone permits of a close fit to the inner tube surface, and when the cleaner is pushed through the tube the thin cutting edge of the cone passes between the soot and the tube, cutting the soot free and carrying it forward where it is deposited in the rear connection of the boiler.

What is known as the Detroit tube cleaner is shown in Fig. 6. It cuts off the soot and carries all loosened refuse

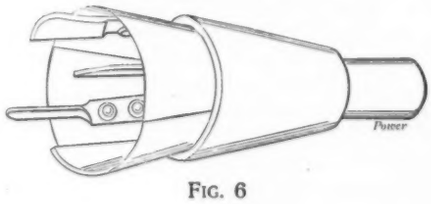


FIG. 6

back into the combustion chamber. The size of the head is adjusted by turning the handle to the right or left. The device can be connected to a steam supply and used as a blower if desired. The elasticity of the spring blades permit a close fit of the blades to the internal surface of the tube, thus preventing the steam blowing out at the front end of the tube. The cleaner consists of a brass cone, three steel blades and springs and a malleable-iron cone. The projecting

fingers permit of easy entrance to the tube. It is made by J. T. Wing & Co.

A different design of scraper is shown in Fig. 7. It consists of four scraping blades hinged at the inner end. Adjusting screws are provided so that a proper fit in the tubes may be obtained. The disk at the base assists in cleaning the

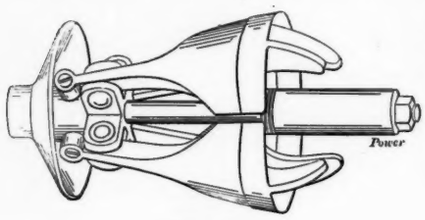


FIG. 7

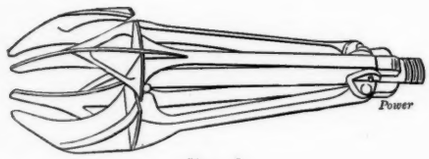


FIG. 8

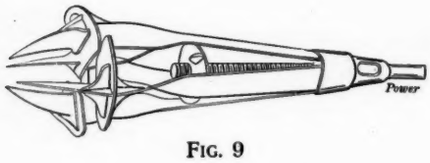


FIG. 9

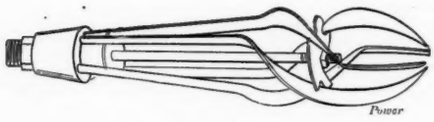


FIG. 10

tubes of loose soot. It is made by Zilliox, Greene & Co.

The scraper shown in Fig. 8 is made up of four scraping blades, secured to arms that are hinged at the inner end, as shown. Suitable springs keep the scraper arms in an out position, but allow the malleable-iron head to easily enter the tube. The scraper blades are made with a shearing plough which cut the accumulation of soot and other deposits from the tube. It was made by the American Steam Packing Company twenty years ago.

The steel tube cleaner, made by the H. W. Johns-Manville Company, is shown in Fig. 9. It is automatic in action, simple and light. The scraper blades are so constructed that they always have a bearing on the inner surface of the tube. The scraping head is made in sections, so that the cutting blades, while free to give for any irregularity in the tube, will fit the surface throughout. The spring arms keep the scrapers expanded, and the tapered outside ends allow the scraper to easily enter the tube. An adjusting nut is shown on the center rod.

A tube scraper, used for years, is shown in Fig. 10. It consists of a cylindrical head, containing four spring scraper blades, together with a central rod on which an outer end-supporting pin is secured. The ends of the scraper blade are so formed that entrance to the tubes is easily accomplished. The scraping edges are bent at about 45 degrees with the spring portion of the blades, and so curved that one overlapped the other, so that the entire surface of the tube is scraped at one operation. This scraper is made by the Jackson Flue Scraper Company.

In Fig. 11 is shown a tube cleaner that was designed some sixteen years ago. The springs were made of rolled tempered steel and were fitted with cast-steel heads

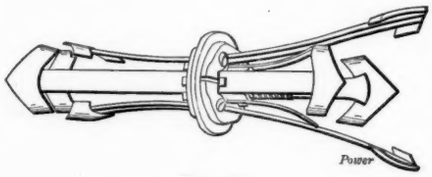


FIG. 11

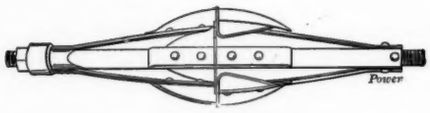


FIG. 12

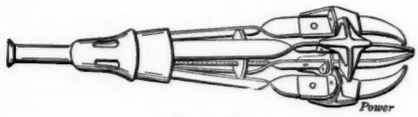


FIG. 13

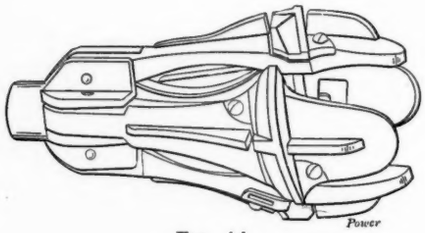


FIG. 14

that could be reground until worn out. The knives had a shear cut, which enabled them to pass over obstructions where a square-cut knife might stick. The small springs forced the front ends of the flat spring together to enable the scraper to enter the tube. After the rear end of the cleaner had entered the tube, both ends of the scraper operated to clean the tube, the leather center pushing or drawing out the soot.

With the design of tube scraper, shown in Fig. 12, the scraping knives are placed in the center, as shown. Each is riveted to a spring that extends from one end of the scraper to the other. On each side of the scraping blades is a curved metal ridge which assists the head to enter the tube. This scraper is made by the Jarecki Manufacturing Company.

Fig. 13 shows an early design of cleaner. The scraping blades are riveted to springs on one side and to a solid piece on the head end, the four pieces forming a cone-shaped head. This permits of the cleaner easily entering the tubes.

Still another design is shown in Fig. 14. It consists of four scraping knives which are hinged at the base. Springs keep the scrapers out against the tube surface.

The cleaner illustrated in Fig. 15 consists of four steel scraper blades, each being secured to a spring which is secured at both ends of the head. Owing to the projecting lug on each scraper knife they are prevented from expanding beyond the tube diameter. When the head is pushed into a tube the outer scraping blades are forced in, and the lugs, acting on the opposite end of the scrapers, permit an easy entrance of the head to the tube. The scraping blades are so arranged that the entire surface of the tube is acted upon each time the cleaner is pushed through.

Fig. 16 shows the tube scraper made by A. A. Hull. It has a spindle on which are fitted the scraper arms and cone-shaped disks. The scraping plates consist of four half disks hung loosely on a square central body of malleable iron. Forged-steel springs are fitted under the

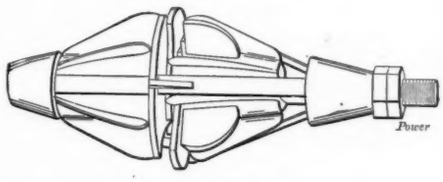


FIG. 15

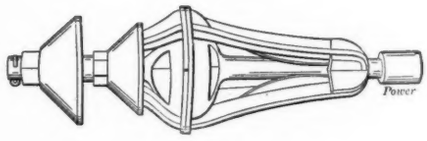


FIG. 16

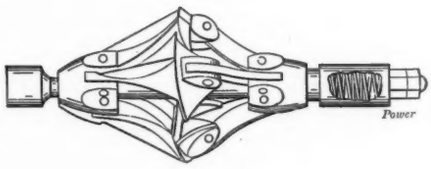


FIG. 17

scraper arms, as shown. The scraping disks are case-hardened to prevent rapid wear.

One type of scraper, made by James L. Robertson & Sons, and several other manufacturers, is illustrated in Fig. 17. It has four scraping blades arranged as shown. The blades are forced out against the surface of the tubes by the spring shown in the end casing. Owing to the gradual curvature of the end links, the scraper readily enters a tube end.

The design of cleaner shown in Fig. 18 consists of four cutting knives which adapt themselves to varying thicknesses of a boiler tube. When the scrapers are forced past an obstruction the spring at the end is compressed by the collar and connections forcing it into the protecting sleeve. It is made by the J. W. Paxson Company and others.

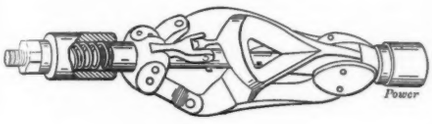


FIG. 18

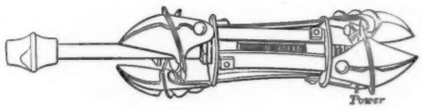


FIG. 19

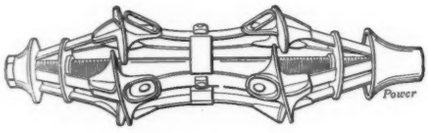


FIG. 20

An adjustable cleaner, made by the Michigan Lubricator Company, is shown in Fig. 19. It has a double scraping head and hardened-steel scraping knives. The adjustment is accomplished by screwing in the center piece which, owing to the collar, will force the adjusting plates toward the center, thus expanding the scraping blades.

What is known as a duplex cleaner is illustrated in Fig. 20. The scraper is fitted with two sets of scraping blades, as shown, so that it may do double duty. The adjustment is made on the center rod which is threaded at both ends. The scraping blades are placed at the center of the springs and are not easily broken off. It is made by the Sherwood Manufacturing Company.

Another design of tube cleaner is shown in Fig. 21. It can be adjusted while in the tube to make a close fit, or made smaller if necessary to pass over uneven surfaces. The adjustment is accomplished by turning the handle rod to which the cleaner is attached. The double heads contain the scraping edges, the latter being riveted to steel springs. A center rod is provided with a right- and left-hand thread on which the two expanded plates are screwed. As both plates move toward or away from the center of the scraper at the same time when turning the center rod, the adjustment of the scraper blades is very sensitive. The center of each spring is bolted to the center piece. Turning the expanded piece toward the center of the cleaner forces the springs out, also the steel scraper tips, thus increasing the diameter of the cleaner. A reverse motion reduces the diameter of the cleaner. It is made by A. W. Chesterton & Co. and others.

The Pratt elastic tube scraper is shown in Fig. 22. It is made by the Millers Falls Company. It has two end caps in which are set the scraping knives. These knives are flexible, but are forced out by means of springs which bear on the underside and at the center, as shown. Owing to the shape of the blades the head has two scraping ends.

An early type of cleaner is shown in Fig. 23. It is self-adjusting and made without springs or nuts. The scraper blades are adjusted by the movement of the rod while the head is in the tube. A central rod has a collar permanently fastened at each end. A loose slotted

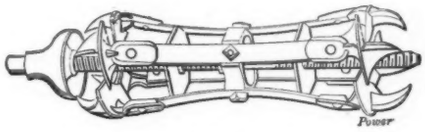


FIG. 21

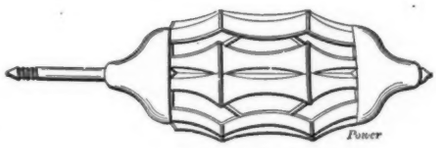


FIG. 22

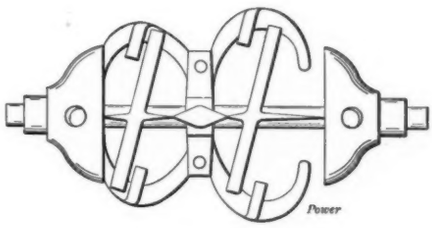


FIG. 23

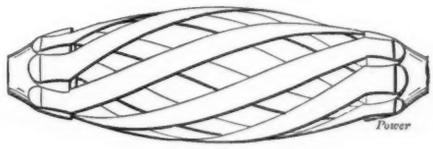


FIG. 24

collar is placed on the rod between the bell collars to which the arms, with scrapers attached, are fastened. When the scraper is pushed into a tube the collar next to the handle becomes engaged with the convergent ends at the adjacent scraping arms, and the opposite ends of the arms are thrown out against the side of the tube. When the scraper is withdrawn from the tube the action of the arms is reversed by the rod sliding through the opposite bell collar.

A modern cleaner of similar design is shown in Fig. 24. The knives are dovetailed on the two heads, as shown, and are so constructed that they are always in contact with the metal surface of the tubes. The knives are self-sharpening. No screws or rivets are used in its construction. It is made by the Scully Steel and Iron Company and others.

An adjustable hand flue cleaner, made by Joseph T. Ryerson & Son, is shown

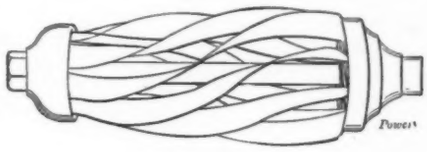


FIG. 25

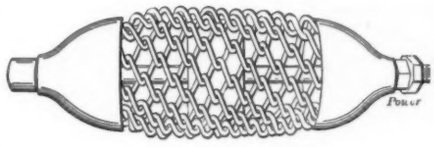


FIG. 26

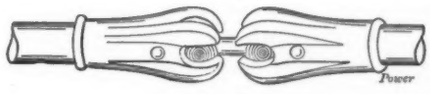


FIG. 27

in Fig. 25. Two nuts on a single bolt are all that are required to hold the parts together. The eight scraping blades are made fast at one end of the cleaner, the other ends being free to move radially inward. This gives the cleaner a degree of adjustability enabling it to meet any ordinary variations in the tube diameter and at the same time thoroughly clean the inner surfaces. The blades are made of spring steel and are so placed as to be easily removable in case it is necessary to make repairs. A similar cleaner is made by the Quincy Flue Cleaner Company.

The "Criss-Cross" cleaner, made by the Reversible Tube Cleaner Company, is illustrated in Fig. 26. It is made of square steel wire which is so woven that it presents to the inner surface of the tube a large number of independent, oblique, serrated cleaning edges, the resilience of which keeps them expanded with considerable force, and yet permits compression in case of blister or other permanent obstruction in the pipe. The cleaner may be adjusted for various sizes of tubes by bringing together or by drawing apart the two ends between which the woven wire is held. The cleaner blades cut when moving in either direction.

A handy ball-joint tube-scraper rod for use in restricted places is shown in Fig. 27. It is made of white ash in sections of various lengths. The sockets and joints are made of malleable iron, the connecting balls of steel. It will fold in any direction and does not kink. It is made by James McCrea & Co.

Fusing Temperatures of Ash*

By E. G. Bailey and and W. B. Calkins

The quantity of clinker formed by a given coal depends upon the composition and the fusing temperature. The effect of the first factor is so little understood that dependence must be placed upon the latter and conditions regulated to suit.

Everyone who is at all acquainted with the burning of coal or coke realizes that the clinker problem is one of very great importance. Probably more shutdowns in boiler plants and steam failures on locomotives are due to the formation of clinkers than to any other one cause. The purchase of coal on specification is often unjust, as no proper method of determining its clinkering properties has been used, and excessive penalties are often exacted for sulphur, due to the erroneous idea that it causes clinker, while other coal commands a premium for high B.t.u. and low ash, but clinkers badly.

The accumulation of clinkers on the grate of any furnace has several bad effects. The most serious one is the reduction of the air supply, which in turn decreases the rate of combustion and the heat-generating capacity of the furnace. In order to obtain maximum efficiency in the burning of coal it is necessary to have the required supply of air distributed uniformly throughout the fuel bed, but the presence of clinkers greatly interferes with this proper distribution of air and causes a reduction in the economy, as well as the capacity, of the plant. Clinkers also add to the labor of firing and greatly increase the expense for repairs to brickwork, grate bars, etc.

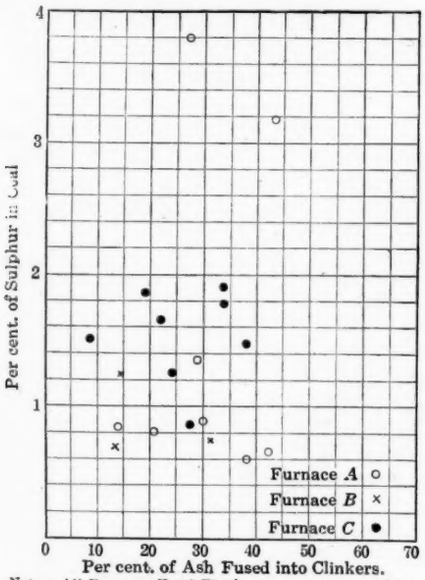


FIG. 1. RELATION BETWEEN SULPHUR CONTENT AND PERCENTAGE OF CLINKER IN ASH

A clinker is a fused mass of ash. The formation of clinker depends upon two factors, the temperature at which the ash will melt and the temperature to which the ash is subjected. It is evident that the temperature to which the ash is subjected is largely dependent upon the method of firing and the rate of combustion and it is often possible to lessen the trouble due to clinkers by changing the method of firing. Even where the furnace conditions are similar, there is a great difference in the quantity and nature of clinker formed from different coals. This difference is directly dependent upon the temperature at which the ash will fuse or melt.

Heretofore, no realible method has been known whereby the clinker-forming properties of a coal could be predetermined, nor has it been possible to make comparisons of this characteristic in different coals. There is a prevalent opinion that the percentage of sulphur is an indication of the trouble which may be expected from clinkers, but this has been proved to be an extremely unreliable and unjust method of comparison. Fig. 1 illustrates very clearly that there is no relation between the percentage of sulphur in a coal and the percentage of its ash which is fused together as clinker. Each of the 19 tests shown was of 24 hours' or more duration and was made on hand-fired boiler furnaces with similar conditions, except the rate of combustion as indicated. All clinkers one inch in size or larger were picked out from the ashes and refuse resulting from the burning of different kinds of coal and the percentage based upon the total ash in the coal burned.

It was impossible to determine the fusing temperature of the ash from all of the above tests, as most of the samples of coal had been destroyed before we were prepared to determine this temperature, but all such data available at the present time are plotted in Fig. 2. The relation here seen to exist between the percentage of the ash that fused into clinkers and the fusing temperature of the ash assures the value and practicability of this determination.

It will be seen that a difference of a few hundred degrees at the higher temperatures has little effect upon the amount of clinker formed, but the percentage of clinker increases very rapidly with a lower fusing temperature of the ash.

An important factor, not brought out by these curves, is that the clinker made from an ash of low fusing temperature will obstruct more grate area than will the same weight of clinker formed from an ash of higher fusing temperature. If this objectionable characteristic could be plotted on the curves of Fig. 2, in addition to the percentage of clinker, the curves would bend toward the horizontal much more abruptly, indicating a still greater increase in the difficulties connected with the burning of a coal with an ash of low fusing temperature.

The percentage of ash must also be taken into consideration, as coal containing 9 per cent. of ash, 20 per cent. of which formed clinker, might be just as objectionable as one containing only 6 per cent. of ash, 30 per cent. of which formed clinker.

Another interesting point brought out by Fig. 2 is the difference between the results obtained in the furnaces *A* and *B*, which is almost entirely due to the difference in the rates of combustion. The coal burned per square foot of grate during the different tests on furnace *A* varied from 19 to 26, with an average of 22 pounds per hour; while in furnace *B* it averaged only 11 pounds per hour and varied from 10.6 to 11.8 pounds. The average temperature in furnace *A* was higher than in *B* on account of the higher rate of combustion, and there would naturally be a greater percentage

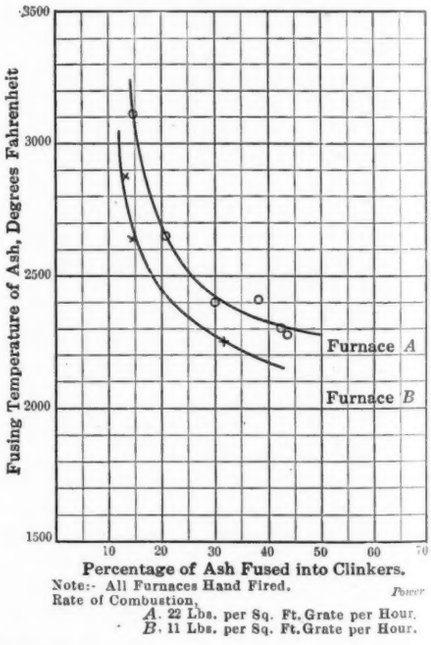


FIG. 2. RELATION BETWEEN TEMPERATURE AND PERCENTAGE OF CLINKER IN ASH

*Abstract from *Bulletin* No. 3, issued by the Fuel Testing Company, Boston, Mass.

of clinker formed from ash of the same fusing temperature.

Many people who have attempted to ascertain the clinkering properties of different fuels have realized the uncertainty of sulphur as an indicator and, due to their inability to determine the fusing temperature of the ash, have depended on other characteristics. The color of the unfused ash and the percentage of iron in the ash have been more generally used for this purpose, but a little study of the accompanying table indicates that no definite relation exists, and that these characteristics are as misleading as is the percentage of sulphur.

Even the complete analysis of the ash, giving the content of silica, alumina and the oxides of iron, lime, magnesia, potash and soda, has often been made with a view to obtaining some knowledge of the fusing temperature. But these constituents, singly or collectively, do not hold a known relation to fusibility, and the importance of the relation of each constituent to all others makes the problem so complicated that only a vague and general idea can possibly be obtained from such an analysis. Doctor Seger, the eminent German scientist, has shown that the fusing temperature of such material may be lowered by adding some flux, such as lime, up to a certain point, while a still greater percentage of it will increase the fusing temperature. In like manner, additional silica may raise or lower the fusing temperature, depending upon its relation to the other constituents, as well as their relation to each other.

Even if one were able to estimate the clinkering property of a coal from the complete analysis of the ash, this analysis alone is much more laborious and expensive than is the direct method of determining the temperature.

TABLE OF COAL CHARACTERISTICS.

No.	Character of Coal.	Fusing Temperature of Ash, Degrees Fahrenheit.	Percentage of			Color of Unfused Pulverized Ash.	
			Ash in Coal.	Sulphur in Coal.	Iron Oxide in Ash.	Relative Density of Color.*	Color Shade.
1	Anthracite	3000	16.11	0.75	4.1	2	pink
2	Semibituminous	2910	7.97	0.96	7.0	10	brown
3	Semibituminous	2880	7.33	0.70	10.0	1	gray
4	Semibituminous	2640	8.90	1.25	12.0	4	pink
5	Gas	2640	9.84	0.72	6.6	3	yellow
6	Semibituminous	2550	12.85	3.10	23.7	16	brown
7	Gas	2520	7.22	1.22	13.0	8	yellow
8	Anthracite	2520	10.08	0.61	6.1	6	yellow
9	Gas	2460	6.91	2.35	24.7	15	brown
10	Gas	2350	7.44	1.98	17.5	12	red
11	Semibituminous	2320	13.06	0.92	7.1	5	yellow
12	Semibituminous	2320	10.76	2.22	20.6	13	red
13	Gas	2300	6.78	0.98	10.6	7	yellow
14	Gas	2260	7.25	3.12	34.0	14	red
15	Semibituminous	2250	7.11	0.73	12.9	11	yellow
16	Coke	2190	10.22	1.40	13.9	9	yellow

*Graded according to density; No. 1, lightest; 16, darkest.

Exhaust Steam Regenerators

By Charles H. Smoot

Regenerators are divided into two general types. In one, the water passes through the incoming steam. The energy required to circulate the water in such regenerators is so great that it considerably lowers the efficiency. In the other type, the steam passes through the water; circulation is rapid, and the mixture of steam and water most thorough.

The exhaust-steam regenerator is a heat accumulator whose function is to equalize variable fluxes of steam, receiving a variable flux and delivering a constant one. When the flux of steam received by the regenerator exceeds that delivered, the function of the regenerator is to absorb the excess, holding it in reserve until a future period occurs when the steam received is less than that required, the deficiency being made up by withdrawing from the regenerator steam which has previously been stored.

When the flux of steam received from the engines by the regenerator is just equal to that taken by the turbines, the pressure within the regenerator is maintained constant, no steam is condensed and the regenerator floats idle. As soon, however, as steam comes to the regenerator in greater quantities than required by the turbine, the pressure in the regenerator is slightly raised, with an increased temperature of steam, which, becoming hotter than the water, allows the water to absorb the excess.

Should the quantity of the incoming steam be less than that required by the turbine, the pressure in the regenerator falls, lowering the temperature of the steam, and allowing the water to give off steam, removing the heat which was previously absorbed during a time of excess.

The regenerator is a storage reservoir, which is floating continually on the system, and it neither absorbs nor gives off steam when the engine exhaust equals the turbine requirements. But as soon as the engine exhaust varies from the turbine requirements, the regenerator comes into play and it absorbs the excess steam or delivers the deficient steam.

That the regenerator may have its entire heat-storage capacity available to absorb a sudden large excess of steam, such as occurs in steel-mill practice, the fundamental necessity is that a sufficient amount of water be always present in the zone of condensation. If the regenerator is so arranged that the entire mass of water is not always present in the zone of condensation, a sudden excess does not come in contact with the entire mass of water and consequently the full absorption capacity of the regenerator is not available to absorb excess steam.

In practice, the conditions of absorption are far more severe than those of delivery, for when the regenerator is called upon to absorb steam it generally happens that the steam to be absorbed is entering at a rate far in excess of the average value of flow to the turbines, so that the maximum flux of steam which the regenerator must absorb is considerably in excess of the maximum rate of flow which the regenerator is obliged to give off.

The steam delivery from the regenerator seldom exceeds the average rate of engine exhaust, which is equal to the maximum steam consumption of the turbine. This marked difference between the capacity for absorption and that for delivery is well illustrated by the accompanying curves, based on actual tests of a large rail mill. The tests show that the length of time during which a regenerator is absorbing steam is only a small portion of the total time of operation, and that when the regenerator is absorbing steam it must absorb fluxes whose rate of flow is far in excess of the rate of flow for steam delivery. This point cannot be too strongly emphasized, for it is here that the steam regenerator as a steam economizer becomes either a failure or a success.

The practical type of regenerator as at present in use employs water as a heat-storage medium, the steam being stored in the water as heat. When the steam flow is in excess, the pressure within the regenerator increases and therefore the steam temperature also increases. If the regenerator is well designed, the entire mass of water being in intimate mixture with the steam, the temperature of the water is obliged to increase with that of the steam; consequently the water absorbs heat from the steam, condensing portions of it and reducing the pressure.

The closeness with which the temperature of the water follows variations in the temperature of the steam depends upon how thoroughly mixed the two elements are in the zone of condensation, for if insufficient water is present to condense the surplus steam flowing at any instant, the variations in pressure are large and the temperature difference between steam and water is excessive. Since a temperature difference of 2.8 degrees Fahrenheit at atmospheric pressure corresponds to one pound per square inch pressure difference, the importance of small temperature variations is apparent.

It is possible, theoretically, to store heat in metal or any other substance which is capable of absorbing heat when its temperature rises. Water, however, is the most effective of all substances for absorbing heat, since it has the highest heat capacity per unit of weight.

Steam regenerators are usually installed in connection with low-pressure steam turbines, although their use is not limited to turbines alone and they may be applied wherever it is desired to change from a variable flux of steam to a continuous flux, or *vice versa*. Regenerators equalize the flow of steam in the same manner as a water reservoir equalizes a flow of water. When used in conjunction with engines whose exhausts vary from time to time, the regenerator, by absorbing the excess steam for future use, saves the future use of an equal amount of high-pressure steam, which, if a regenerator were not present, would be required to make up the future deficiency in the exhaust-steam supply.

Application of Regenerators in Rolling-mill Practice

Fig. 1 is an illustration of a typical steam flux which regenerators are called upon to equalize in a rail mill. The engines whose exhausts comprise the steam received by the regenerators consist of one blooming-mill, one roughing-mill and two finishing-mills, all of which engines work more or less in conjunction, since they act upon the same steel in its passage through the mill, beginning as a bloom and ending a finished rail.

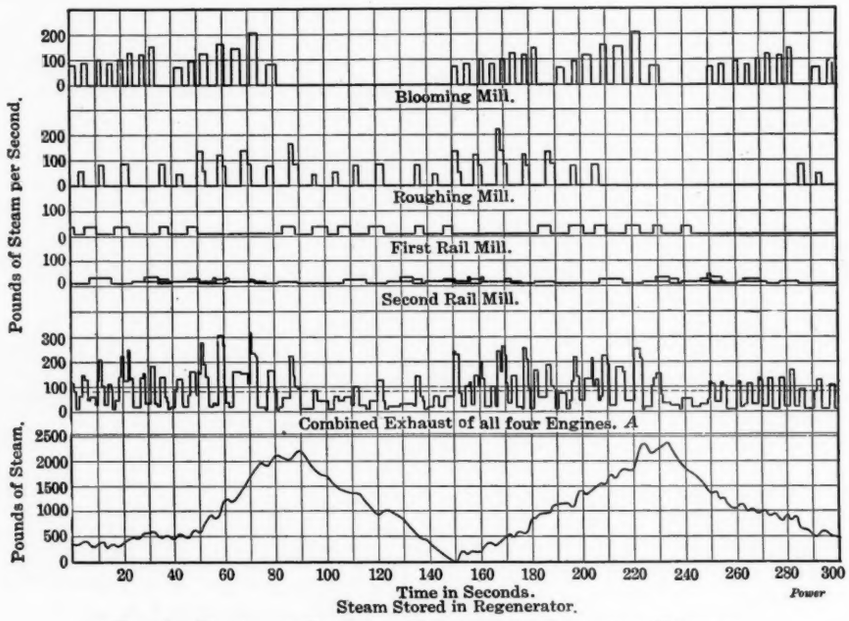


Fig. 1. Diagrams Showing Exhaust of Rail-mill Engines

These curves were obtained for the purpose of investigating the advisability of a regenerator plant. The abscissas or horizontal distances represent time in seconds. The ordinates or vertical distances represent quantities of steam.

Curve *A* represents the flux of steam in pounds per second delivered by all four engines working under normal conditions. The steam flow is extremely variable, having small rises and falls occurring within four- or five-second intervals, as well as much larger periods of excess and deficiency, whose intervals are of about one minute duration.

The dash line passing through this curve at the hight for 80 pounds per second shows the average steam flow from the four engines. Wherever the line of total steam flux passes above the dash line, the flux is in excess of the turbine requirements and some of it must, therefore, be absorbed by the regenerator. The feature which is most strikingly apparent is that the rate of steam flow in excess of the average value reaches maximums which are far in excess of the rates of flow needed to make up deficiencies, thus illustrating the fact that the regenerator is more severely taxed during the periods of steam absorption than during the periods of steam delivery.

The lowest curve, more or less sinusoidal in shape, shows the total weight of steam in pounds which must be stored in the regenerator at the corresponding instant. Where this curve is rising, the steam from the engine exceeds the flow to the turbine, and, conversely, where it is drooping, the regenerator, by giving off steam, is making up the deficiency of steam supplied by the engines.

The average time between crests and troughs is something like one minute; consequently, the approximate period during which the regenerator is storing an excess or supplying a deficiency is of one minute duration.

The greatest total amount of steam stored in the regenerator occurs at 233 seconds from the start, at which time the regenerator contains 2350 pounds of steam more than at 150 seconds; at the latter point the reading is zero.

A well designed low-pressure turbine installation employing this amount of steam, viz., 80 pounds per second, or 290,000 pounds per hour at 28 pounds per kilowatt, is capable of generating 10,000 kilowatts.

To apply a regenerator to this typical rail mill requires that the regenerator be capable of passing to the turbines an average steam flow of 80 pounds per second, and that it be capable of absorbing between the greatest and lowest pressures permissible in the regenerator (usually atmosphere and four pounds above) 2350 pounds of steam.

The amount of water theoretically necessary to absorb this quantity of steam equals:

$$\frac{2350 \times \textit{average latent heat}}{\textit{Temp. at 4 lb. gage} - \textit{temp. at 0 lb. gage}} =$$

$$\frac{2350 \times 962}{12.4} = 182{,}000 \textit{ pounds}$$

The economy effected by the regenerator is easily estimated. The omission of the regenerator would oblige all surplus steam over the 80-pound rate to discharge to the atmosphere. The total weight of steam which would be so discharged during the time interval covered by the curve *A* in Fig. 1 amounts to 8800 pounds in the 300-second period. This is equivalent to a continuous loss of 106,000 pounds per hour.

In addition to this waste of exhaust steam, in the periods of deficiency the required amount would have to be supplied from the high-pressure mains. The total amount supplied during the time

covered by the curve is equivalent to about 80,000 pounds an hour.

The steam taken by the turbine, amounting to 80 pounds per second, is equivalent to 290,000 pounds per hour. Consequently, the loss of steam due to the absence of a regenerator in such an installation causes a waste of

$$\frac{186,000}{290,000} = 0.64$$

or 64 per cent. of the average steam taken by the turbines.

Absorption

As affecting the regenerator design, usefulness and efficiency, the consideration of prime importance is the question of steam absorption and not that of steam delivery. Any mass of water will give off steam when its pressure is reduced below the critical point, or the point at which the temperature is equal to the temperature of saturated steam at the same pressure. On the other hand, the absorption of steam is of vital importance, since all of the important energy losses in the regenerator occur during this part of its action, which losses, however, can be fully controlled by the design of the apparatus.

The efficiency of the regenerator as a piece of power apparatus is, as usual, the ratio of power output to power input.

If the regenerator delivers steam at the same pressure as that at which it receives steam, barring the very slight losses due to radiation, its efficiency is 100 per cent. On the other hand, if the regenerator always receives steam at a higher pressure than that at which it can deliver steam, its efficiency is markedly less than 100 per cent., and the loss is represented by the pressure drop between the absorbing condition and the condition of delivery. This depends entirely upon the regenerator design and especially upon the provision for maintaining within the condensing zone a large volume of water in proportion to the flux of steam.

If the flux of water is 276 pounds per pound of steam, the temperature difference between the steam which is being condensed and the water doing the condensing will be not less than 3½ degrees Fahrenheit, or 1¼ pounds per square inch pressure difference. The pressure difference demonstrates itself at the instant when the regenerator changes from an absorbing action to a delivery action, since the pressure must fall 1¼ pounds. These figures apply to a highly efficient regenerator.

A machine which provides a flux of water through the condensing zone less than 276 pounds per pound of steam is proportionately less efficient. If the flux of water is reduced to, say, 20 pounds per pound of steam, the temperature difference between the steam and the water becomes approximately 48 degrees, corresponding to a pressure drop from 19 pounds absolute to 7 pounds absolute, or a loss in pressure of 12 pounds.

In that type of regenerator in which steam and water are mixed in a separate condensing vessel located below the main regenerator reservoir, and in which water passes from the upper vessel by the action of gravity into the lower vessel, where it is sprayed in small streams into the incoming steam flux, practical considerations of construction limit the flux of water to not over 40 pounds per pound of steam; and since 2.8 degrees Fahrenheit difference in temperature corresponds to one pound per square inch difference in pressure, this type of regenerator cannot give off steam until the pressure is reduced by at least

$$\frac{962}{40} = 24$$

degrees Fahrenheit, or

$$\frac{24}{2.8} = 8.6$$

pounds per square inch in the loss of pressure between the conditions of absorption and those for delivery. Since the total pressure range available to the turbine between the admission pressure and the pressure in the condenser does not exceed some 13 pounds per square inch, the above loss in pressure due to the insufficient absorption capacity of the regenerator amounts to 57 per cent. in available pressure at the turbine.

In order that a regenerator may be capable of absorbing excess steam as it occurs in usual rolling-mill practice, it is necessary that an intimate mixture of steam and water be continually preserved and that the temperature of the water follow closely the temperature of the steam as it rises or falls with corresponding variations in pressure. This requires that a large mass of water be always in contact with the steam flux. The only appreciable losses of energy which occur in a regenerator arise from a failure in this respect.

The temperature of the steam during the process of absorption must always be, unavoidably, slightly greater than the temperature during the evolution of steam. The design of the regenerator, however, can greatly reduce the resulting loss of energy, and it is this consideration which led Professor Rateau, the originator of the regenerator principle, to abandon long ago the idea of regenerators in which water is caused to flow through steam, substituting for this arrangement that in which steam is caused to flow through water.

The first arrangement is incapable of absorbing large quantities of steam in short periods, because it is impracticable to force the circulation of the enormous volume of water required to condense the steam when the temperature difference between the steam and the water is limited to less than 5 degrees Fahrenheit, which is the limit in the design adopted for all the Rateau regenerators and which corresponds to about 1.8 pounds per square inch pressure drop between absorption and delivery of steam.

To work within a limit as close as this requires a circulation of 195 pounds of water for each pound of steam, since the latent heat of steam is 962 B.t.u. per pound in the neighborhood of atmospheric pressure, and to absorb the heat contained by one pound of steam by water whose temperature increase is limited to 5 degrees Fahrenheit requires, therefore, about 195 pounds of water.

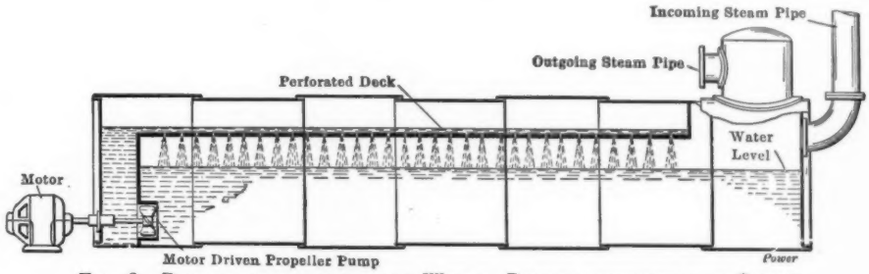


FIG. 2. REGENERATOR IN WHICH WATER PASSES THROUGH THE STEAM

The enormous flux of water required by practical regenerators is, therefore, obvious. Where steam is to be handled in quantities as great as 300,000 pounds per hour, the circulation of water through the condensing zone of the regenerator must be 195 times this, or 585,000,000 pounds per hour. To pump such an amount of water against a head of even 10 feet is an obvious impossibility, but in the Rateau regenerator fluxes of water as great as this are readily realized, since the water circulation has a very large cross-sectional area and the length of the path is reduced to only a few feet. The circulation occurs around a guide plate and is maintained by the passage of steam in very small bubbles through the upward rising current of water.

Any attempt to operate a regenerator with a forced circulation of water, in which the water is sprayed into the steam, must inevitably prove a failure in so far as economy of operation is concerned, for the reasons just outlined.

The designer of a regenerator acting on this principle is confronted with the following dilemma: In order that the regenerator may be capable of absorbing steam efficiently, i.e., with a small temperature difference between absorbing water and steam to be absorbed, the water circulation through the condensing vessel must be in the neighborhood of 195

pounds of water per pound of steam. To pump this quantity of water requires a large amount of power, which is lost, impairing the efficiency. On the other hand, if to economize the power required to maintain the water circulation, the quantity of water is reduced, the regenerator has its absorption capacity reduced and becomes incapable of absorbing the full excess of steam as it occurs, and necessitates equipping the regenerator with a relief valve on the turbine side, which will blow to atmosphere a large portion of the excess incoming steam.

A regenerator which is well designed will absorb steam fluxes which may be

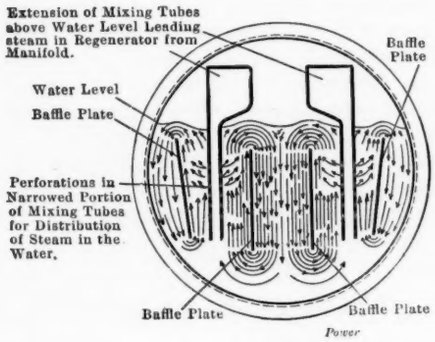

Extension of Mixing Tubes above Water Level Leading steam in Regenerator from Manifold.
Baffle Plate
Water Level
Baffle Plate
Perforations in Narrowed Portion of Mixing Tubes for Distribution of Steam in the Water.
Baffle Plate
Baffle Plate
Power

FIG. 3. REGENERATOR IN WHICH STEAM PASSES THROUGH THE WATER

five times as great for short intervals as the flux of steam continuously taken from the regenerator by the turbines, but this can be accomplished only by a regenerator which is so designed that the entire mass of water is brought into the zone of condensation many times in each minute. A regenerator in which the water is forced to circulate through a special condensing chamber separate and distinct from the main regenerator reservoir, for the reasons outlined above, is incapable of absorbing an excess even as great as the average flux of steam required for the turbine without discharging steam to atmosphere.

The practicability of designing a regenerator along these lines, which will efficiently absorb large fluxes of steam, can be readily grasped by comparing it with a barometric condenser, whose cooling water is not more than 5 degrees Fahrenheit colder than the steam to be condensed. It takes a very effective condenser to heat the condensing water up to within five degrees of the temperature of the incoming steam; consequently, a regenerator which acts along similar lines is obliged to be more effective as a condenser than the best of barometric condensers at present in use, since a temperature difference of five degrees between water and steam will correspond to a very appreciable loss of energy in a regenerator.

In constructing regenerators of new types, the feature of prime importance is this question of steam absorption; the question of steam delivery is a matter which takes care of itself.

RELIEF VALVES

In addition to the regenerator proper, the steam valves which are called on to pass variable fluxes of steam encountered in steel-mill work are of great importance. The usual relief valves as at present designed for service with condensing engines are wholly inadequate for this work. The passage of steam occurs with the violence of an explosion, and the valve must be capable of opening and closing its full travel many times in each minute without pounding itself to pieces.

Some of the valves required are over 36 inches in diameter, with moving parts whose weights may reach as high as 1000 pounds, and must oscillate up and down with a travel of some 6 inches or more, 25 or 30 times a minute. The mechanical difficulties of constructing such valves are apparent. The energy of the blow with which such a valve would strike its seat would be on a par with that of a steam hammer if special provision were not made to absorb the energy of the blow. This violence of action arising from the rapidly changing flux of steam given off by blooming mills and similar engines absolutely prohibits the use of any moving parts in a regenerator which are subject to this action, since such parts would very rapidly wear themselves to pieces unless the rapidity of their action were restricted; and if this is done, then the free passage of steam into the regenerator becomes restricted, and large back pressures are placed on the engines.

TYPES OF REGENERATORS

To classify the different types of regenerator construction is somewhat difficult as numerous forms have been built and proposed. The principal difference, however, between the proposed methods of construction consists in either causing the exhaust steam to flow through the water contained in the regenerator shell, or causing the water which is to absorb the steam to flow in finely divided streams through the steam.

Fig. 2 illustrates diagrammatically an effective type of regenerator, in which the water passes through the steam. The motor-driven pump acts against a head of some 5 feet, lifting the water from the main reservoir to the upper chamber, from which it passes in a very large number of fine streams through the steam to be condensed. The action of this regenerator to absorb steam is the same as the usual barometric condenser, only a far larger space must be provided in which to mix the steam and water. The great difference between this regenerator and a condenser, however, lies in the fact that the temperature of the water which is to condense the steam must be nearly the same as that of the steam itself; consequently far more water is required than for a condenser.

If a very large volume of water is not kept in circulation, the absorption capacity of this type of regenerator becomes limited, and the loss of energy becomes large.

On the other hand, the type illustrated in Fig. 3, in which the steam is caused to pass through the water, allows the entire mass of water in the regenerator to be intimately and rapidly mixed with the incoming steam. The circulation of water is maintained by the passage of steam in very small bubbles upward in the rising current of water, causing a velocity of flow which brings all of the water into the zone of condensation at least once every three seconds. This feature gives to this type of regenerator a vastly greater rate of steam absorption than can be obtained in the type illustrated in Fig. 2.

With reference to the giving off of steam which has been absorbed, these two types of regenerator are very much alike. There is one exception, however, which gives considerable preference to the type shown in Fig. 3. This lies in the fact that the passage of the steam through the water maintains a circulation, as indicated by the arrows. This circulation brings to the surface all of the water at regular intervals, where its pressure is reduced to the lowest value existing within the regenerator shell. Since all of the water within the regenerator shell is kept at the same temperature, a reduction in pressure of steam above the water allows the surface water to give off sufficient steam to cool it from the maximum temperature to the new temperature determined by the new pressure. On the other hand, the water lying at the bottom of the regenerator cannot begin to give off any steam until the pressure of the steam over the water has been reduced by an amount somewhat greater than the pressure corresponding to the hight of water above that at the bottom.

In other words, if the water in a regenerator is uniformly at the temperature of 212 degrees, corresponding to atmospheric pressure, and if the depth of the water is 6 feet, then a reduction in steam pressure of 2.62 pounds per square inch will just bring the pressure of the water at the depth of 6 feet to atmospheric pressure, at which point it begins to give off steam.

The water which was at the surface of the regenerator, however, begins to give off steam immediately a reduction in steam pressure occurs. If no circulation exists, the water lying at the bottom of the shell will take very little part in the action of giving off steam, unless the reduction of pressure above the water is excessive. This difficulty can be entirely overcome by giving the water a continuous circulation, bringing to the surface at regular intervals all of the water con-

tained within the shell, so that during the process of giving off steam all of the water within the regenerator is brought to the surface, where the static head is removed, and the water is allowed to give off its full share of steam.

The type of regenerator illustrated in Fig. 2 cannot be provided with any very practical means for keeping the large mass of water circulating in the main reservoir, and therefore the steam given off by the water in the lower portions of the vessel will be very much less than for the upper portions, thus reducing the effective capacity of the regenerator as a whole.

An interesting comparison of these two types of regenerator can be made by applying them to the steam flux illustrated for the rail mill in Fig. 1. The maximum flow of steam to the regenerators occurs at the instant 60 seconds, and amounts to 300 pounds per second. In order that no exhaust steam will be wasted, the absorption capacity of the regenerators for this installation should be equal to this maximum rate of flow. Taking the pressure variation permissible in the regenerators to be four pounds per square inch above atmosphere, the maximum and minimum temperatures of steam corresponding to these two pressures will be 224.4 degrees at four pounds above and 42 degrees at atmospheric pressure. The latent heat of steam within this pressure range is 962 B.t.u. per pound; consequently, to condense one pound of steam,

$$\frac{962}{224.4 - 212} = 77.5$$

pounds of water will be required.

The total weight of water required within the shells, on the basis that each pound of water is capable of giving off its full share of steam is

$$77.5 \times 2350 = 182{,}000 \text{ pounds.}$$

In a regenerator of the type of construction illustrated in Fig. 3, the full amount of water is brought where it may act upon the incoming steam once every three seconds, giving, therefore, a flux of water for the purpose of condensing steam which amounts to 60,700 pounds per second.

The maximum flow of steam which the regenerator must absorb is

$$300 - 80 = 220$$

pounds per second, which requires

$$220 \times 77.5 = 17{,}000$$

pounds of water per second. Therefore, this type of regenerator is capable of absorbing a steam flux far in excess of that which actually occurs. In order that the regenerator illustrated in Fig. 2 may be capable of absorbing this same steam flux, the flow of water through it must be 17,000 pounds per second, and if the effective head through which this water must circulate be taken at 10 feet, the power expended in the circulating pump will be

$$\frac{17{,}000 \times 60 \times 10}{33{,}000} = 308$$

water horsepower, which, with an overall efficiency for pump and motor of 40 per cent., will require an input of 770 horsepower.

Another type of regenerator, similar to that illustrated in Fig. 2, since in it a flow of water is introduced in such a way as to pass through the incoming steam, has recently been constructed in this country, and tests are soon to be made. This apparatus is illustrated diagrammatically in Fig. 4. In principle it differs from that illustrated in Fig. 2 through using the impact of incoming steam against condensing water to lift the water from the lower chamber back into the upper chamber, and relies upon the action of gravity to introduce condensing water from the upper into the lower, or condensing vessel. In this apparatus the available head of water for creating a flow from the upper to the lower chamber is approximately 10 feet. The maximum theoretical velocity which water can attain flowing under 10-foot head, with no allowances whatever for frictional resistance or obstructions, is 25.5 feet per second. Consequently, a regenerator of this type, to have a steam-absorbing capacity great enough to absorb the maximum flow of steam (220 pounds per second) given by the engines whose performance is represented on curve *A*, Fig. 1, must be 17,000 pounds per second, which, at the above given velocity, would require that the pipe *S* in Fig. 4 have a sectional area of 10 square feet or a diameter of 3 feet 7 inches.

To properly subdivide a flow of water of this extent will require that the lower chamber *C* be of very large dimensions, and it is somewhat doubtful if the incoming steam can entrain and deliver to the upper chamber so great a flow of water, whose weight in pounds per unit of time is

$$\frac{17{,}000}{300} = 56.6$$

times the weight of incoming steam for the same interval.

To use the steam flow for the purpose of lifting this water in place of the usual motor-driven pump is an advantage, since it eliminates some auxiliary apparatus, but on the other hand, the efficiency of the motor-driven pump for lifting water is very much greater than the efficiency of the incoming steam to do the same thing, since the incoming steam lifts the water in a similar manner to the use of steam in an injector for boiler feed, the efficiency of which when well designed for the purpose cannot exceed 20 per cent. The efficiency of the steam in this type of regenerator for lifting water must be far less than this, since in designing the steam and water chamber, no attempt has been made to conform to the lines of the boiler injector.

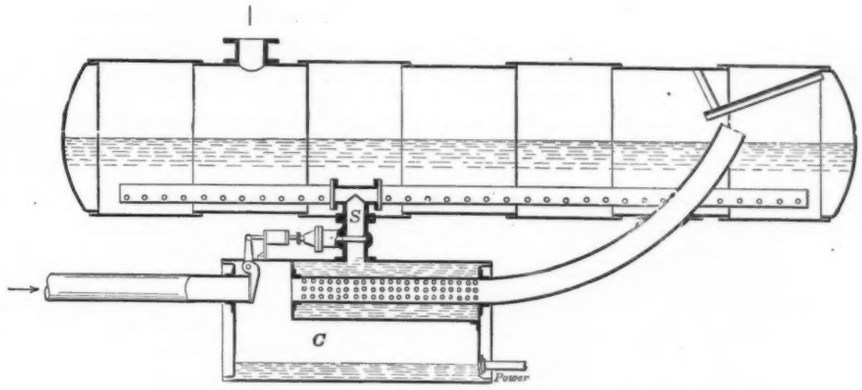


FIG. 4. REGENERATOR BASED ON THE INJECTOR PRINCIPLE

The first plant equipped with a steam regenerator was designed and put in operation by Prof. A. Rateau at the Bruay mines, France, in 1890, the source of exhaust steam being a hoisting engine.

In our country the first installation was that of the Wisconsin Steel Company, South Chicago, where the regenerator receives the exhaust steam of a reversing blooming engine and delivers steam to a low-pressure turbine of some 600 kilowatts capacity.

A second plant was installed at the Vandergrift works of the American Sheet and Tin Plate Company the following year; the regenerator handles the exhaust of a reversing mill engine and supplies steam to a Rateau-Smoot low-pressure turbine of 600 kilowatts capacity.

The Rateau Steam Regenerator Company has installed eight plants with capacity ranging from 500 to 2000 kilowatts.

In Europe, plants aggregating a capacity of 400,000 kilowatts are operating in conjunction with Rateau steam regenerators.

Old Hal Mossback struck his firm fer a raise tother day an' he got it. Th' Old Man kicked him so dumd hard thet he raised 'bout six feet frum th' floor of the engine room.

Electrical Department

Induction Motor Hand Starters

By R. H. Fenkhausen

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Starters for Large Motors

For motors of 75 horsepower and above, some type of starter giving several starting positions is desirable in order that the line disturbance may be reduced. Fig. 14 shows a starter of this class having five starting points. The lever is moved progressively from the "off" position through the several starting positions (exactly as in a direct-current starting box) to the running position, where it is locked against backward movement. To stop the motor it is necessary only to move the lever one notch further in the same direction as when starting, which carries it to the "off" position. The internal connections of this starter are shown in Fig. 15. The contacts for each phase or circuit division are mounted on a separate drum, and to prevent short-circuiting successive taps of the auto-transformers, the contacts of one tap are opened before those of the next are closed. The drums are so geared together that only one phase of the motor winding is open-circuited at any given moment, so that at least one phase of the motor is receiving current at any time, which prevents a drop in speed between starting points in case the operating handle is moved too slowly from point to point.

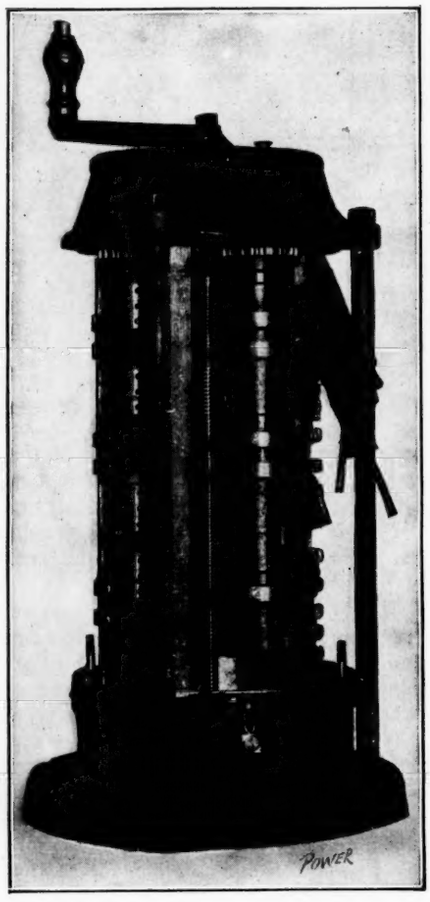


Fig. 14. Drum and Finger Type of Starter

For 2200-volt motors the circuit is never opened between successive points, and to limit the current which would necessarily flow if two autotransformer taps were short-circuited, a "preventive" resistance is automatically inserted in the circuit between each pair of taps just before they are connected.

Starters of the type shown in Fig. 14 are built with the switch contacts submerged in oil contained in a sheet-iron tank. To gain access to the contacts the oil tank must be lowered, which is accomplished by shifting a clutch and turning the operating handle backward. It is essential, however, that a pit be provided below the starter into which the tank may be lowered. The reason for this will be apparent from the following incident. An engineer once installed a starter of this type and poured in the oil at the filling plug provided at the top, without giving the removability of the oil tank a thought. The starter was used in connection with a motor driving an important pump, and when one morning some of the fingers failed to make contact with the drum, it was impossible to gain access to them because there was not room to lower the tank; consequently, the pump could not be started. Luckily plenty of help was available, so with the writer's assistance a pit was hastily dug, the base of the starter being supported by a couple of timbers across the excavation, and the tank was lowered. New fingers were soon in place and the pump was running twenty-five minutes after the telephone call for assistance had been received. You may be sure that pits were immediately provided under all the starters of this type in the plant. The best method to employ is to sink a short length of oil-well casing beneath the starter just large enough to clear the tank, and then fill the bottom of the casing and the space surrounding it with concrete, in the top of which the foundation bolts for the starter base may be placed.

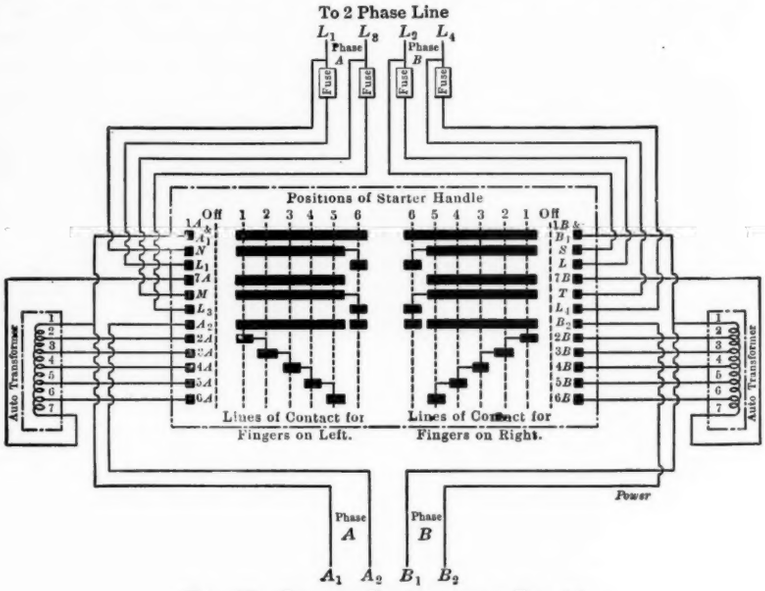


Fig. 15. Circuit Diagram for Fig. 14

Motor Containing its Starter

Constant-speed induction motors with wound rotors and the starting resistance mounted on the rotor spider are self-contained and require no external starting device other than the main switch. The starting lever on the motor frame should, if possible, be mechanically interlocked with the main switch in some way that will prevent a careless operator from closing the main switch when the starting handle on the motor is not in the

starting position.* A motor of this type is illustrated in Fig. 16.

Protective Devices

The hand controllers now built for induction motors do not embody any mechanism for overload or no-voltage release. The overload protection is obtained as a rule by the use of fuses or small circuit-breakers separate from the starter proper. No-voltage release is, however, a feature of some of the oil-break circuit-breakers now made, and it is valuable to large motors, because if the power supply should be interrupted it will prevent the motor from starting under full voltage when the power is again turned on.

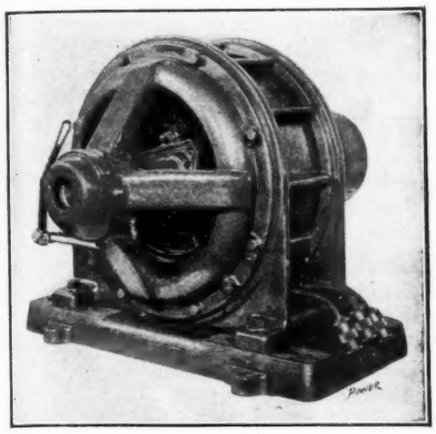


Fig. 16. Motor with Starter in Rotor Spider

When selecting fuse blocks for the protection of an induction motor, the National Underwriters' Code rating of fuses should be considered. The size of fuse-block changes for every few capacities and the range of fuses suitable for each size of block is given below. A different size of block is used also for 250 and 600 volts.

1 = 0 to 30 amperes, ferrule contacts.

2 = 35 to 60 amperes, ferrule contacts.

3 = 65 to 100 amperes, knifeblade contacts.

4 = 100 to 200 amperes, knifeblade contacts.

5 = 200 to 400 amperes, knifeblade contacts.

If, for instance, a 100-ampere fuse is required, a 100- to 200-ampere block should be installed with a 105-ampere fuse. If the starting current of the motor is greater than estimated it is a simple matter to insert a larger fuse, but if a 100-ampere block were installed a larger fuse could not be used and the fuse blocks would have to be changed.

*One way of doing this was described in my article on page 807 of the issue of May 3, 1910.

Primer of Electricity

By Cecil P. Poole

Dynamo Field Excitation

In the lessons of November 16 and December 7, 1909, and January 4 of this year the principles of the electromagnet were explained. The student should "brush up" on these principles now because the field magnet of every dynamo (excepting a few special types of very small output) is a big electromagnet, and the principles explained in the lessons mentioned apply to these big magnets exactly as they do to small ones.

The field magnet of a dynamo is given that name because it creates and maintains the magnetic field through which the armature conductors must pass in order to generate electromotive force. Many writers use the term "field" indiscriminately to mean "field magnet," "field-magnet winding," "field-magnet coil," etc.; the word has only the one meaning, however, of a space occupied by magnetic lines of force: the magnetic field. The magnetic field of a dynamo is always located in the short spaces between the magnet polefaces and the armature, appropriately known as the "air-gaps," and around the edges of each poleface.

In most cases the field magnet of a dynamo is magnetized by current supplied by its own armature; such machines are called "self-exciting." As may be imagined from the sense in which it is used, "exciting" is another name for "magnetizing"; it is a much more convenient term to use, however, as will be evident as we go along and the various ways of using it develop. Moreover, there is a certain distinction between "excitation" and "magnetization"; an electromagnet is "excited" by passing a current through its winding, and in this case "excited" also means "magnetized"; but a piece of steel may be magnetized by rubbing it against a powerful magnet, and, although it is "magnetized," it is not "excited." From this it is evident that to excite really means to magnetize by means of a current passing through a winding surrounding the core of the "excited" magnet.

Of course, the current with which an electromagnet is excited may be taken from any suitable source; when the field magnet of a dynamo is excited by current supplied from some other source than its own armature, it is known as "separately excited."

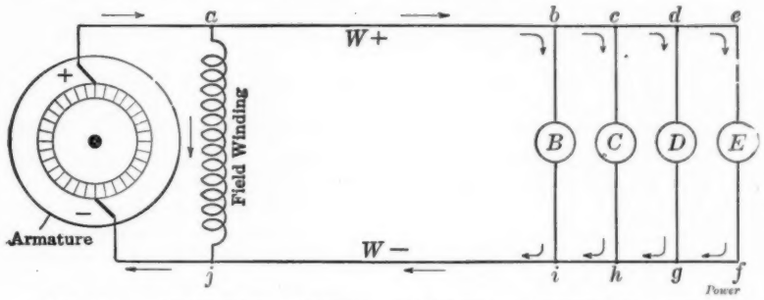


Fig. 70. Elementary Diagram of Constant Potential System

Field-magnet windings of self-excited dynamos are divided into two general classes: one called "shunt" windings and the other called "series" windings. If the student remembers the early lessons on parallel and series circuits, he will understand from these two names that the shunt field winding is connected in parallel with some other circuit and the series field winding in series with some other circuit; in both cases this "other circuit" is the external or work circuit supplied by the dynamo. Fig. 70 is an elementary diagram of the connections of a shunt-wound dynamo and Fig. 71 that of a series-wound dynamo. As the diagrams indicate, the load of a shunt-wound dynamo consists of a number of receiving devices (represented in the diagram by circles) connected in parallel with each other, and the field winding is connected in parallel with these devices, while the load supplied by the series-wound dynamo consists of devices

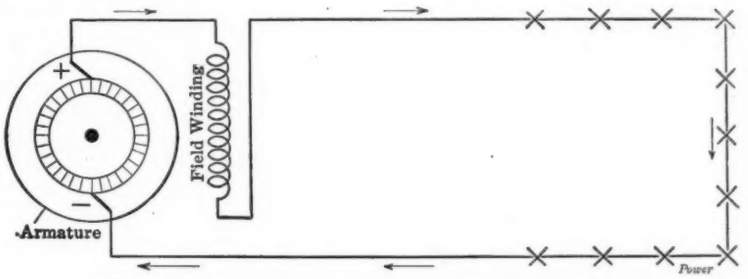


Fig. 71. Elementary Diagram of Series System

(represented in this diagram by crosses) connected in series with each other and with the field winding of the dynamo.

The Shunt-wound Dynamo

The shunt-wound machine is used more generally than the series-wound machine, for reasons which will appear as the student goes along. Referring to Fig. 70, the arrows indicate the direction in which the current passes through the various paths in the system. Starting at the positive brush of the dynamo (marked +) the full armature current passes to the point *a*, where a small portion of it splits off to go through the field winding; the remainder goes on to the point *b*, where another portion leaves the main wire and passes through the device *B*, indicated by the circle; this happens successively at the points *c* and *d*, and at *e* what is left of the original current goes through the device *E* to the other wire of the circuit at *f*, and joins the current from *D* at *g*; at *h* the combined currents from *f* and *g* join the current from *C*, and at *i* the current in the main wire *W*— is further augmented by the current from *B*. Between the points *i* and *j* the current in this wire is the sum of the currents passing through the four devices, and at *j* it is increased to the full armature current by the small current coming through the field winding.

If the main wires *W*+ and *W*— to which the load is connected be regarded for the moment as having no resistance, it will be evident that the electrical pressure, or difference of potential, from *a* to *j* is exactly the same as that from *b* to *i*, that from *c* to *h*, and so on; in other words, the potential between the two main wires *W*+ and *W*— is the same at all points of those wires, and is equal to the electromotive force of the armature. For this reason, a parallel circuit supplied by a shunt-wound dynamo is called a "constant-potential" circuit.

When the armature of a shunt-wound dynamo is driven at a constant rate of speed, if the resistance of the field winding remains unchanged, the voltage or difference of potential at the brushes of the machine remains almost constant no matter what the load may be; if the armature winding had no resistance, the voltage would remain absolutely constant. The reason for this is that the voltage of a dynamo is determined by the speed of the armature and the strength of the magnetic field; the strength of the field depends entirely on the current passing through the field winding, and this depends on the voltage at the terminals of the winding and the resistance of the winding. It is obvious, then, that so long as the resistance of the field winding is not changed the current in the winding depends on the volts at its terminals, and if the armature speed is not changed, there is nothing to change the voltage. In practice, the voltage at the brushes of a shunt-wound dynamo drops slightly as the load (and consequently the armature current) increases, because the resistance of the armature winding and brushes uses up more and more of the generated voltage as the armature current increases, leaving less and less voltage for the outside circuit. Because of the property just described, of maintaining almost a constant voltage under all loads, the shunt-wound dynamo is frequently called a "constant-potential" dynamo, although it is not strictly one.

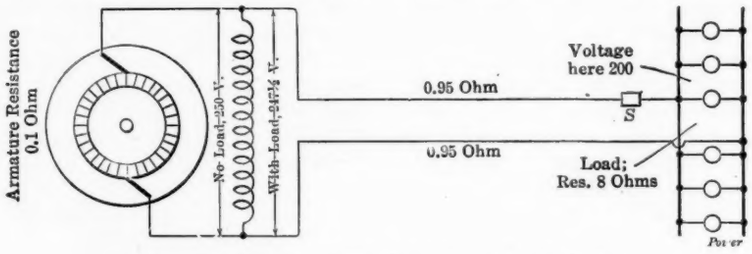


Fig. 72. Illustrating Effect of Line Drop

The current that passes through the field winding of a shunt-wound dynamo is a small proportion of the total armature current at full load; in small machines it may be as much as 15 or 20 per cent. of the full-load armature current, but in machines of ordinary and large sizes it is usually from 10 to 2 per cent. of the total current, according to the size of the dynamo. Hence, the field winding constitutes a shunt or bypass to the main or load circuit, and that accounts for the name "shunt winding."

Because of the fact that circuit wires have resistance, and therefore cause a "drop" in voltage between the dynamo and the receiving devices, it is necessary to adjust the voltage of shunt-wound dynamos instead of allowing them to maintain a practically constant voltage at the brushes. Reference to Fig. 72 will assist in making this clear. Suppose that the dynamo, when running at normal speed and with the load cut off by opening the switch *S*, generates 250 volts and that the resistances of the various parts of the system are as stated on the diagram. Now, suppose the switch *S* to be closed; the dynamo will be sending current through a total of 10 ohms (ignoring the field winding), and the current therefore will be

$$\frac{250}{10} = 25 \text{ amperes}$$

The voltage used up in driving the current through the armature would be

$$25 \times 0.1 = 2\tfrac{1}{2} \text{ volts},$$

and that used up in driving the current through the two line wires would be

$$1.9 \times 25 = 47\tfrac{1}{2} \text{ volts},$$

making a total loss of 50 volts in the armature and the main-circuit wires. This leaves only 200 volts at the load, which should receive 250 volts. If the load should be increased until its resistance is only 3 ohms instead of 8,* the total resistance of the whole system would be $3 + 1.9 + 0.1 = 5$ ohms, and the current then would be

$$\frac{250}{5} = 50 \text{ amperes}$$

The drop in the armature and brushes would be

$$50 \times 0.1 = 5 \text{ volts}$$

and that in the main-circuit wires would be

$$50 \times 1.9 = 95 \text{ volts},$$

making a total of 100 volts. This would leave only 150 volts at the load.

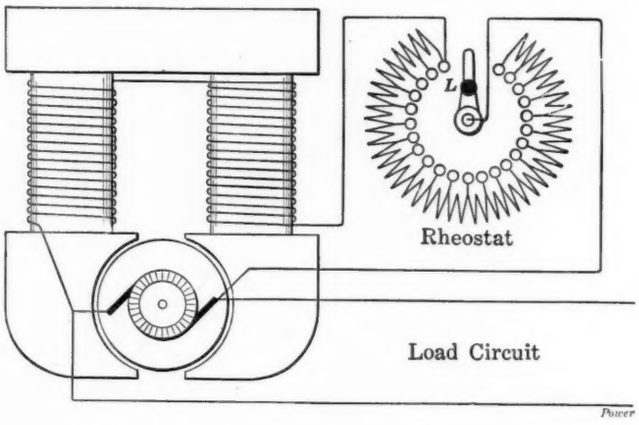


Fig. 73. Shunt-wound Dynamo Connections

The assumed conditions just cited were made much worse than they would be in actual work in order to emphasize the disadvantage of voltage drop in the connecting wires of a circuit, but under real operating conditions with the full load connected the drop might easily be 10

*Increasing the load in a parallel circuit is done by adding more devices in parallel, which reduces the joint resistance of the total load.

per cent. of the voltage that the dynamo would give with no load at all; that is, the voltage at the load might be 225 volts instead of 250.

In order to correct this disadvantage, the field winding of the dynamo is made "stronger" than would be required to produce the rated voltage of the machine, and an adjustable resistance, known as a "rheostat," is inserted in the field-winding circuit, as represented diagrammatically in Fig. 73. The zig-zag line arranged in a circle represents a high-resistance conductor connected at intervals to contact buttons; a lever *L* is pivoted at the center of the circle and arranged so that its free end will come into contact with the buttons when the lever is swung around the pivot.

As it is shown in the diagram, the lever *L* does not touch any of the buttons, and the circuit of the field winding is therefore open; if the lever be moved to touch the first button on the right, the entire resistance will be connected in series with the field winding; moving the lever to the next button will leave out the first section of the resistance conductor and therefore the resistance of the whole field circuit will be reduced. Each "step" of travel of the rheostat lever puts an additional section of resistance out of circuit, until the last button is reached, when the rheostat conductor is all cut out and the current in the field winding is maximum. The resistance of the regulating rheostat is usually two or three times as great as the resistance of the field winding, and the range of field-strength adjustment is therefore ample for all ordinary conditions.

Stability and Maximum Voltage

If the magnetism of a dynamo field magnet increased directly in proportion to the strength of current passed through the field winding a shunt-wound dynamo would be very unstable—that is, its electromotive force would not tend to stay at any given value; if it happened to increase, the current in the field winding would increase proportionately, this would increase the field strength proportionately, and that would cause another increase in the electromotive force, which would increase the exciting current again, and so on, until the voltage rose so high as to destroy the insulation of the machine. Fortunately, when the magnetic density in iron or steel reaches a fairly high value (about 65,000 lines per square inch in the best grades of steel or wrought iron) increasing the exciting current does not produce a proportionate increase in magnetic flux. Therefore, when a dynamo is running at normal speed, if its field circuit be closed it will begin to generate a feeble electromotive force and this will increase rapidly until the magnetic density in the iron and steel reaches such a point that a large increase in voltage at the terminals of the field winding will produce only a small increase in magnetic flux; the electromotive force cannot increase any further automatically because the requisite increase in magnetic flux cannot be caused by the increase

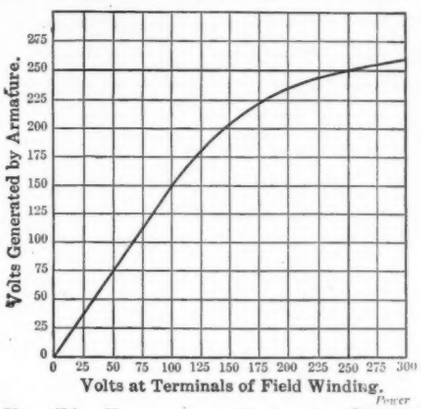


FIG. 74. EXCITATION CURVE OF SHUNT-WOUND DYNAMO

in exciting current that it itself would produce.

For example, suppose that in order to produce enough magnetic flux to generate 250 volts in the armature, an exciting current of 5 amperes were required in the field winding and the resistance of the winding were 50 ohms; then the voltage at the terminals of the winding would have to be

$$50 \times 5 = 250 \text{ volts},$$

or exactly the same as the generated voltage. Now suppose also that in order to generate 251 volts the exciting current had to be increased to 5.1 amperes; as the resistance of the field winding is 50 ohms, a voltage of

$$50 \times 5.1 = 255$$

would be required, and as the armature would only give 251 volts with this additional excitation, it would be impossible for it to supply the 255 required, and the voltage would remain at 250.

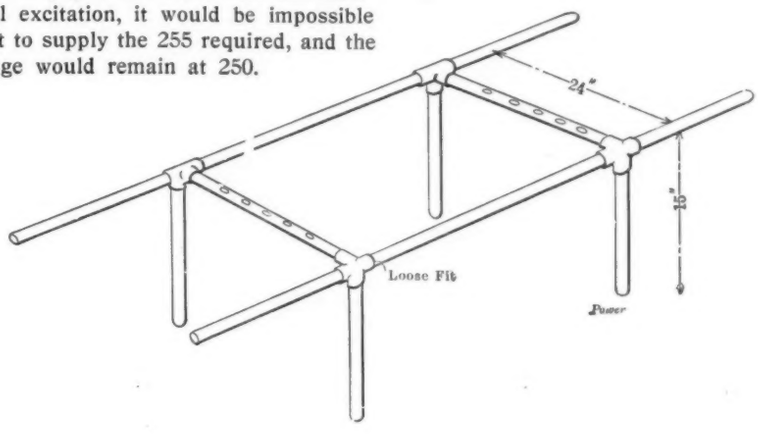
This effect may be better understood, perhaps, by reference to Fig. 74, which is an excitation curve for an actual dynamo rated at 230 volts. Starting at any number on the scale of generated volts and tracing over to the curve and thence down to the scale at the bottom will show the number of volts required at the terminals of the field winding to make the armature generate the number of volts at which the tracing began. For example, in the machine tested, to generate 125 volts it was necessary to apply 81.3 volts at the field-winding terminals; a voltage of 162 at the field-winding terminals caused the armature to produce 212.5 volts, and so on, *the generated volts being higher than the exciting volts* at all points of the curve until 250 was reached; then the two voltages were the same, and as any greater generated voltage required more exciting volts than the armature could generate, the machine could not produce more than 250 volts when operated self-exciting. The test from which this curve was plotted was made by exciting the field magnet with current taken from another circuit instead of from its own armature; by doing this, the test could be carried beyond the ability of the machine. To generate 260 volts in the armature required 300 volts at the terminals of the field winding.

From the foregoing it will be evident that *the maximum possible voltage of any shunt-wound self-exciting dynamo is that which is required at the terminals of the field winding to produce sufficient magnetic flux to enable the armature to generate the same voltage.*

LETTERS

An Armature Stretcher

The accompanying sketch illustrates a sketch used for carrying damaged armatures to and from the shop. Although it is used for this purpose more than any other, it is handy also for carrying other things, such as short shafting. The



AN ARMATURE CARRYING "STRETCHER"

sketch shows the construction. The legs and cross pieces are made of 1¼-inch pipe and the handles of 1-inch pipe. The handles fit loosely in the crosses so that the pairs of legs may be set at any distance apart. The ⅜-inch holes in the cross pieces are for the insertion of pins to keep the armature from rolling.

J. J. O'BRIEN.

Buffalo, N. Y.

Gas Power Department

Elementary Lectures on the Gas Producer

By Cecil P. Poole

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Producer Gas

What is commonly called producer gas is really a mixture of several gases, as the reader might suppose from looking at Fig. 12 and reading the third lecture of this series.* As the incandescent zone of the generator makes carbonic-acid gas and the decomposition or second zone makes carbonic oxide and hydrogen the gases leaving the generator must include these three, unless all the carbon dioxide (CO_2) is used up in the second zone in the production of carbonic oxide (CO), which is practically impossible. Then, again, the air that supports combustion in the fire zone contains nitrogen, which will not burn or combine with any of the other gases present in a generator, and this, too, passes out through the delivery pipe.

The exact composition of the delivered gases depends largely on the kind of coal used in the generator. The soft coals all contain considerable quantities of hydrocarbons which are readily distilled out in the top zone of the generator, and the gas made by such coals therefore contains these hydrocarbons. They are marsh gas, consisting of one part carbon and four parts hydrogen, and olefiant gases containing two parts of hydrogen to one of carbon, by volume. There are also some hydrocarbons that are not "fixed" when they are distilled out, but condense after they get away from the heat of the generator, forming tar, and some which are "broken up" by excessive heat in the upper part of the generator, the hydrogen passing off as a separate gas and the carbon being carried off in the form of fine dust, commonly called lampblack.

The hard coals (anthracite and semi-anthracite) contain very little of hydrocarbons and what there are are fixed gases, that is, they do not condense into tar when cooled. Consequently, the gas made from hard coals contains only a small proportion of hydrocarbons, no tarry matter and no lampblack. The percentage of hydrogen in anthracite gas is also slightly less than in gas made from bituminous coals and lignites, partly because the coal itself contains less and partly because less steam is required in the air supply to keep the temperature of the fuel bed within the desired limits.

The proportion of free hydrogen in the delivered gas depends largely on the amount of steam admitted to the fuel bed, per pound of coal "gasified," and the steam supply depends on the character of the fuel and the type of generator

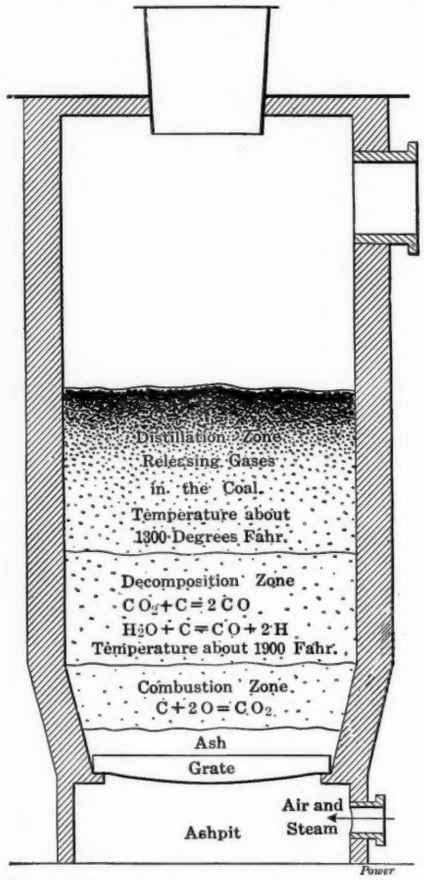


Fig. 12. Approximate Character of the Zones of a Fuel Bed When Steam Is Admitted with the Air

used. Roughly, under average conditions, the total amount of steam passed through the fuel bed, including that produced from the moisture in the coal, should not be less than 6 per cent. nor more than 10 per cent. of the weight of air passed through the fuel bed in the same length of time. In estimating the composition and heat contents on the delivered gas, it is customary to assume that one-fourth of the oxygen used in forming CO_2 and CO is supplied by steam and three-fourths by air, and the amount of steam admitted from the vaporizer is less, of course, when the coal contains much moisture than when it is fairly dry. For example, a hard coal containing 4 per cent. of moisture would require twice as much steam from the vaporizer as a soft coal containing 8 per cent. of moisture, per pound of carbon gasified, and a lignite containing 16 per cent. of moisture would need only half as much steam per pound of carbon gasified as the soft coal with 8 per cent. of moisture, *provided the percentage of fixed carbon were the same in all three.* This, of course, is an absurd proviso, because the percentage of fixed carbon in anthracite cannot possibly be the same as that in bituminous coal, and the percentage in the latter is always more than in lignite.

To get at a real comparison of the steam required by different grades of coal, therefore, both the amount of fixed carbon and the amount of moisture contained in the coals must be considered. Thus, if an anthracite coal contained 80 per cent. of fixed carbon and 4 per cent. of moisture, and if one-tenth of the carbon were burned to CO_2 and nine-tenths used to make CO, the process would work out as follows for 100 pounds of coal:

8 pounds carbon burned to CO_2 require	18⅔ lbs. oxygen
72 pounds carbon burned to CO require	96 lbs. oxygen
Total	114⅔ lbs. oxygen
One-fourth of the oxygen supplied by water	28⅔ lbs. oxygen
Total weight of water required to supply oxygen = 28⅔ × 9 ÷ 8 =	32⅛ lbs.
Moisture in the coal	4 lbs.
Steam required to be supplied by vaporizer	28⅛ lbs.
Oxygen to be supplied in the air	86 lbs.
Air required	345⅛ lbs.

Proportion of steam in the air supply = 28.25 ÷ 345.25 = 0.0818, or about 8.2 per cent.

If a grade of bituminous coal containing 69 per cent. of fixed carbon and 4 per cent. of moisture be considered on the same basis, it will be found that 24½ pounds of steam are required from the vaporizer for each 100 pounds of coal, as compared with 28¼ for the anthracite having the same percentage of moisture; and considering a grade of lignite to be used in which there are 36 per cent. of fixed carbon and 15 per cent. of moisture, it will be found that *no steam at all* is required from the vaporizer, because the moisture in the lignite is sufficient to make all the steam needed.

These examples, it must be remembered, are based on the assumed proportions of one-fourth oxygen supplied in steam and three-fourths in air, but there is nothing hard and fast about this divi-

*August 9, 1910.

sion of responsibility; the air may supply as much as 80 or 85 per cent. of the oxygen and as little as 65 or 70 per cent. The proportions assumed were taken because they are a fair average, and some of oxygen thus obtained there must also pass through the generator about 3.3 pounds of nitrogen, air being composed of about 76 parts nitrogen to about 23 of oxygen, by weight (the proportions of oxygen and nitrogen in air have never been determined precisely).

TABLE 6. GAS FROM LIGNITES.

	Composition.	Colorado Lignite.	Texas Lignite.	Peruvian Lignite.
Fuel	Fixed carbon	41.65	29.76	31.00
	Volatile matter	32.26	29.25	38.40
	Moisture	20.24	33.71	18.75
	Sulphur	0.60	0.53	3.25
	Ash	5.65	6.75	8.60
Gas	Carbon monoxide	17.38	18.22	22.15
	Hydrogen	11.05	9.63	9.67
	Hydrocarbons	5.00	4.81	2.06
	Carbon dioxide	10.12	9.61	6.47
	Nitrogen	56.45	57.73	59.65

definite proportions had to be assumed before any estimate could be made of the steam supply.

While it is impracticable to draw any positive comparisons between the results obtained with hard coals and those given by soft coals and lignites, because of the wide variation in operating conditions and, consequently, the wide differences between results obtained with exactly the same coal, it is interesting and instructive to consider some representative performances of producers using different grades of fuel. Table 6 gives the analyses of gases made from three kinds of lignite, together with the proximate analyses of the lignites themselves. The composition of the gas is given by volume, which is customary, but the composition of the fuel is by weight, as also customary.

TABLE 7. GASES FROM HARD AND SOFT COALS.

Constituents.	Bituminous.	Anthracite.
Carbon monoxide	22%	26%
Hydrogen	18%	15%
Hydrocarbons*	3%	2%
Carbon dioxide	4%	7%
Nitrogen	53%	50%

*Not counting tar.

It will be noticed that all three of the gas analyses show more than half of the delivered gas to be nitrogen. This is almost always the case, no matter what kind of coal is used, because the largest part of the oxygen required for combustion and making CO is supplied in the air, as already explained, and for each pound delivered by the generator depends, of course, on the proportions of the various

There is not as much difference between the composition of the gas made from bituminous coals and that made from hard coals as there is between the lignite gases and others, except as to the tarry matter and lampblack already mentioned. These are not included in the gas analysis, however, because they are not gaseous when cooled. Table 7 gives specimen analyses of gases made from hard and soft coals. Of course, these are not to be considered as a definite comparison of the two kinds of gas, for the reasons previously given.

Heat Value of Producer Gas

The number of heat units contained in a pound or a cubic foot of producer gas varies considerably. Gas made from lignite and soft coals contains from 140 to 160 heat units per cubic foot (the temperature being assumed to be 32 degrees, Fahrenheit); that made from anthracite coal ranges from about 120 to 145 heat units per cubic foot. The higher heat value of the gas made from soft coals is due to the greater proportion of hydrocarbons and the slightly greater percentage of hydrogen usually in the gas.

The heat value of the composite gas* combustible gases contained in the composite gas. As in the case of coal, the heat value is the number of heat units liberated by combustion, but it is common practice to figure the heat value of a cubic foot of gas instead of a pound, as is done in the case of coal. As with coal, the heat value of a gas or a mixture of gases can be determined more accurately by means of an apparatus known as a calorimeter, but it is frequently desirable to get at an approximate idea of the heat value without going to the trouble of making a physical test. For purposes of study, too, the estimated heat value, derived from an analysis of the gas, is usually the only one that can be made, because the gas being studied exists only on paper.

As gases expand when heated, and the thermal units contained depend ultimately on the weight of each constituent gas, the heat value is usually based on a temperature of 32 degrees Fahrenheit, not because gases are commonly used at any such temperature, but because that is the temperature at which ice melts and it is, therefore, easily determined with exactness. Unless some other temperature is specified, therefore, the reader will understand that all gas heat values are supposed to be those of a cubic foot of gas at 32 degrees.

Before attempting to figure heat values of composite gases, it is necessary to know the heat values of the elementary and compound gases of which they are composed. There are only four combustible gases to be considered in producer work; namely, hydrogen (H), marsh gas or methane (CH_4), ethylene (C_2H_4) and carbon monoxide (CO); in practice, marsh gas and ethylene are not usually considered separately because there is such a small quantity of the latter in the producer gas that it is not worth the trouble to figure on it separately from the marsh gas; the two are ordinarily lumped together under the head of "fixed hydrocarbons."

Table 8 gives the properties of the four combustible gases so far as is necessary in estimating the heat value and weight of a composite gas containing them, and the air required for combustion. From the data in this table the heat value, combustion air and products of combustion may be readily reckoned

TABLE 8. COMBUSTION DATA OF GASES.

At 32° Fahrenheit and Atmospheric Pressure.

Gas.	Symbol.	Pounds per Cubic Foot.	B.t.u. per Cubic Foot.	Oxygen Used.*	Air Used.*	Products of Combustion.			
						Water Vapor.	Carbon Dioxide.	Nitrogen.	Total.
Hydrogen	H	0.005589	347	0.5	2.4	1.0	0.0	1.9	2.9
Marsh Gas	CH_4	0.04464	1072	2.0	9.6	2.0	1.0	7.6	10.6
Ethylene	C_2H_4	0.07809	1713	3.0	14.4	2.0	2.0	11.4	15.4
Carbon Monoxide	CO	0.07807	342	0.5	2.4	0.0	1.0	1.9	2.9

*Cubic feet used for combustion of each cubic foot of gas.

*The expression "composite gas," though somewhat clumsy, is used to keep in the mind of the reader the fact that a producer makes a mixture of gases, uncombined chemically with each other, and not a compound gas, which consists of chemically combined elementary gases.

for any mixture of gases of which the proportions are known. For example, suppose a composite gas made from bituminous coal contained 18 per cent. hydrogen, 2 per cent. marsh gas, 1 per cent. ethylene and 20 per cent. of carbon monoxide; the heat value per cubic foot would be reckoned as follows:

Hydrogen, 0.18 cubic foot at 347 B.t.u. =	62.46
Marsh gas, 0.02 cubic foot at 1072 B.t.u. =	21.44
Ethylene, 0.01 cubic foot at 1713 B.t.u. =	17.13
Carb. monox., 0.2 cubic foot at 342 B.t.u. =	68.40
Total	169.43

As rich a gas as this is seldom obtained in actual practice. The reason for the abnormally high heat value is the assumption of 3 per cent. of hydrocarbons; if these are assumed not to be there, the heat value of the gas immediately falls to 131 B.t.u. per cubic foot, which is a value often obtained.

The air required for the combustion of such a gas would be ascertained as follows:

For the hydrogen	0.18 × 2.4 = 0.432
" " marsh gas	0.02 × 9.6 = 0.192
" " ethylene	0.01 × 14.4 = 0.144
" " carbon monoxide	0.2 × 2.4 = 0.480
Total	1.248

or 1¼ cubic feet of air per cubic foot of gas burned.

Repairing a Broken Crank Shaft

By J. A. Luther

A short time ago the crank shaft on our 10-horsepower engine broke near the cheek, at *A*, Fig. 1, and upon ordering

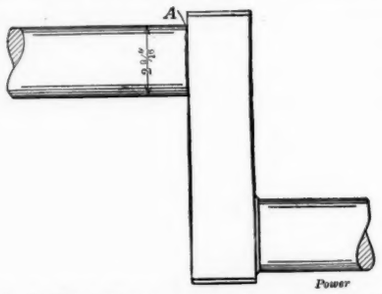


Fig. 1. The Broken Part of the Shaft

a new one we were informed that it would be at least five weeks before one could be shipped. Having a forfeit job on hand that had to be completed immediately, we simply could not afford to wait, and therefore decided to repair the broken shaft in our own factory, although it was not at all an easy repair to make.

Upon examining the broken ends we found that the shaft had an old crack in it, the fresh break being only about ⅞ inch cross-section, and it must have been running for months in that condition.

How the Shaft Was Repaired

The piece of the shaft that was broken off the crank cheek was accurately trued up in a lathe and the broken end was turned to a taper 1½ inches from the end, the taper being ⅛ inch to the foot; see Fig. 2. Next, the crank cheek was drilled with a tapering hole to fit the tapered end of the broken-off shaft, as indicated at *C* in Fig. 3. During the boring process a round piece of brass, fitted tightly in the space, was inserted between the crank cheeks in line with the shaft,

Fig. 2. Tapered End of Broken Shaft

as at *D*, Fig. 3, to avoid springing the cheek being bored, and a weight was attached to the lathe chuck to counterbalance the crank. The entrance to the taper hole *C* in the crank cheek was bored straight for ¾ inch and was made 0.004 inch smaller than the diameter of the shaft, to get pressing fit. The tapered

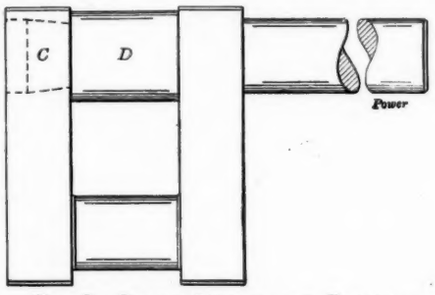


Fig. 3. Crank Braced for Boring Cheek

end of the piece of shaft was pressed home in the crank cheek and the two were pinned together with a ½-inch tapered pin through the center of the shaft (at right angles to the axis), the protruding ends of the pin being heavily upset to prevent working loose.

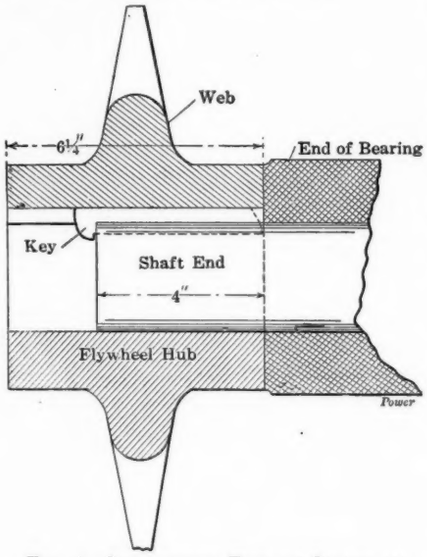


Fig. 4. Shortened End of Shaft in Flywheel Hub

The finished shaft was, of course, about 2¼ inches short on the repaired end because it had been inserted that far in the crank cheek. Consequently, it did not reach all the way through the flywheel hub when placed in position in the bearings, the result being about as indicated in Fig. 4; there was only 4 inches of shaft in the hub, which was 6¼ inches long. We expected this would give trouble, but we drove the key home solidly and started up, trusting largely to good luck.

Contrary to our apprehensions, the wheel on the repaired end of the shaft ran practically true but the one on the other end, where nothing had been changed except to drive out the old key and fit a new one, ran considerably out of true. This was eventually corrected by repeated peening.

The engine has now been running with full load for nearly three months, and apparently is as good as new. The expected trouble of the flywheel working loose on account of its short bearing on the shaft has not occurred, and we have practically stopped looking for any new complications to develop from the repaired crank shaft.

Corrosion of Water-cooled Exhaust Piping

By M. W. Utz

At the plant where I am employed there are two three-cylinder four-stroke-cycle vertical gas engines running on natural gas; each engine runs 12 hours per day, as the plant runs 24 hours a day. We have a small stream of water piped to the exhaust manifold to cool the exhaust gases; as this seemed to attack the exhaust pipe, causing it to waste away in a short time, the pipe was tapped near the exhaust manifold, making a sort of trap for the water from which it was drained by a 1-inch pipe to the sewer.

This drain pipe and fittings were eaten through by the combination of water and exhaust gases in two months, so I changed to galvanized pipe and fittings, which seem to last much longer. Before changing to galvanized pipe and fittings I tried a few galvanized nipples in the line, using black pipe elsewhere, and when the iron pipe was eaten through the galvanized pipe was still in fairly good shape.

If any of the readers of Power have had any similar trouble and overcome it, I wish they would explain how in these columns.

Steam Turbines Superseded by Gas Engines

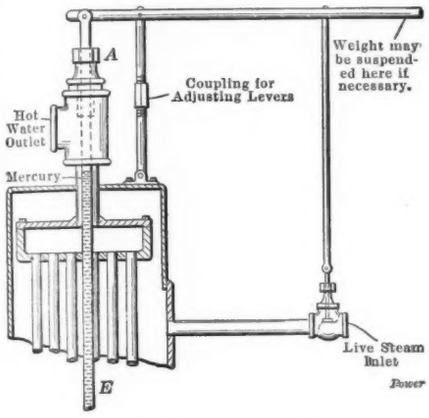
Steam turbines as prime movers at the coal mines of Mitsui & Co., in Japan, says *Gas and Oil Power*, are to be superseded by gas power. Two large 3200-brake horsepower Nürnberg engines to be run on coke-oven gas have been ordered for the generation of electricity at the Müke mines. The steam turbines will be retained as reserve units.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Temperature Regulating Valve

Some months ago a temperature-regulating valve was installed in connection with our heaters. The apparatus is patented, but the accompanying sketch snows some improvements upon the original arrangement. A ¾-inch wrought-iron pipe *E* extends from the top *A* to the bottom of the heater, passing down through the center tube. This pipe is filled with mercury which, in expanding and contracting, regulates the admission of steam through an arrangement of levers. The valve is for live-steam heating only and is entirely automatic.



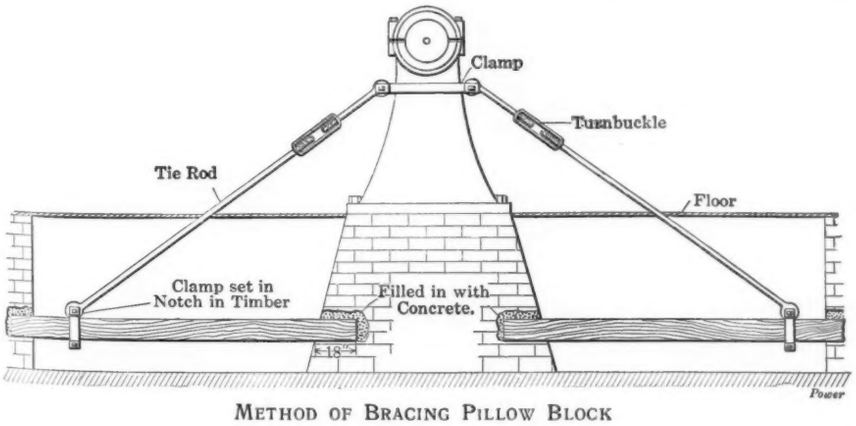
VALVE CONNECTED TO HEATER

JAMES E. NOBLE.

Toronto, Canada.

Repairing a Pillow Block

Several years ago I was engaged to make some repairs at a power plant containing three 300-horsepower tandem-compound engines. The outboard bearings of these engines were giving considerable trouble by vibrating badly, which was due partly to the form of foundation employed and partly to the fact that the plant was built on swampy ground. The accompanying sketch shows how the repairs were accomplished.



METHOD OF BRACING PILLOW BLOCK

A heavy iron clamp was fitted and held by cap screws around the pedestal, close up under the bearing, and to this were attached heavy tie rods, each containing a turnbuckle near its upper end. The lower ends of the tie rods, which also terminated in "eyes," were clamped to timbers in the cellar. These timbers were placed at right angles to the shaft and were held in place by being set in notches in the foundation and walls. The timbers were then blocked up and the notches rammed full of concrete. After the concrete had set, the motion of the pillow blocks was taken up by the turnbuckles on the tie rods and the vibration ceased.

GEORGE LITTLE.

Passaic, N. J.

Repairing Valve Stems

At *A*, Fig. 1, is shown the valve stem in a pump, worn where it passes through the stuffing box. As it was impossible to keep it packed while in this condition, and as the pump was used only during the winter, it being a receiver pump on a heating system, the stem was cleaned where it had worn, and filled in with solder, as shown at *B*. The solder was then filed down to the size of the stem, and the pump ran with this repair for about a year before a new stem was put in.

In Fig. 2 is shown a main-valve stem *G*, and the cutoff-valve stem *F*, concentric with the former. The cutoff-valve stem wore where it passed through the stuffing box on the main-valve stem *G*. The end *E* of the stem screws into the cutoff valve; whereas the ball, of the ball and socket joint, screws on at *C*. By cutting a thread at *C*, the same length as the thread at *E*, reversing the stem and screwing the end *C* into the valve, the stem was made to last just twice as long as usual.

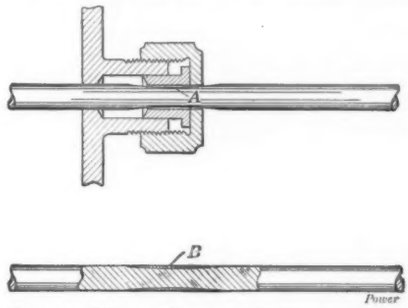


FIG. 1. WORN PORTION FILLED WITH SOLDER

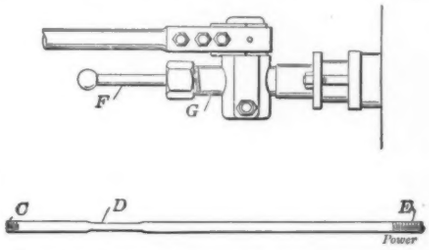


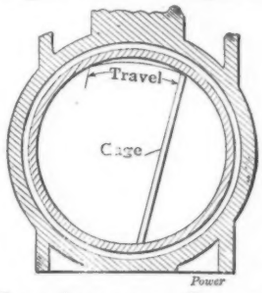
FIG. 2. CUTOFF-VALVE STEM REVERSED

R. L. MOSSMAN.

Tampa, Fla.

Fitting Pistons to Cylinders

The accompanying table for fitting pistons to cylinders was adopted by a large marine-engine firm in England after a series of exhaustive tests, and is applicable to cylinders which have been bored, and, to a certain extent, to cylinders which have not worn too much out of round.



GAGE APPLIED TO CYLINDER

In using this table a wire gage is first made of ¼- or 5/16-inch round iron and of a length equal to the diameter of the cylinder at its smallest section. The ends of the gage are tapered and are

TABLE OF GAGE TRAVELS.

Diameter, Inches.	Gage Travel, Inches.	Diameter, Inches.	Gage Travel, Inches.
1	1/4	63	4
3	1/2	66	4
6	5/8	69	4 1/4
9	11/16	72	4 1/2
12	3/4	75	4 5/8
15	7/8	78	4 3/4
18	1	81	5
21	1 1/4	84	5 1/2
24	1 1/2	87	5 1/2
27	1 3/4	90	5 5/8
30	1 3/4	93	5 3/4
33	2	96	6
36	2 1/4	99	6 1/4
39	2 1/4	102	6 1/2
42	2 1/2	105	6 5/8
45	2 3/4	108	6 3/4
48	3	111	7
51	3 1/4	114	7 1/4
54	3 1/2	117	7 1/4
57	3 3/4	120	7 1/2
60	3 3/4	...	...

further filed to give the proper travel. By travel of the gage is meant the distance one end may be swung when the other end is stationary; see the accompanying sketch. If the piston is made of a diameter equal to the length of the gage, it will bear on almost one-half its circumference and will be comparatively tight against leakage, even if a packing ring is omitted. In making gages it is advisable to make them in duplicate, one for the machinist who is to turn the rings, and the other to be kept by the engineer.

W. E. RECTOR.

Brooklyn, N. Y.

Priming a Centrifugal Pump

We had a centrifugal pump which raised water from the river to a settling tank located about 20 feet above the top of the pump. The latter, which was of the double-suction type, was direct connected to a horizontal engine which ran at 175 revolutions per minute. Each suction branch was 30 inches in diameter with a check and stop valve and was connected at right angles to the main suction line, which was 42 inches in diameter. Two other pumps of the plunger type were connected to a tee at the end of the main suction. The discharge was 24 inches with a stop valve close to the pump, and on top of the pump there was a 4-inch pipe with a valve which was always left open. On top of this valve was a nipple and a tee, from one end of which there extended a horizontal pipe that reached to the corner of the building about 12 feet distant, and from this a riser reached to within a few inches of the ceiling, the top of this pipe being 12 feet higher than the water in the settling basin. From the other end of the tee a pipe containing a valve extended to within a few feet of the sewer. This latter valve was always opened before stopping the engine, as there was usually some air in the pump. If it had not been opened the air would have gone out through the riser to the ceiling, carrying water with it. This riser should have been extended through the roof. There was no siphon on this pump as it was primed with water from the discharge pipe and was under pressure from the settling basin, the check valve on the suction preventing the water from flowing back to the river.

I had been at this plant about a month when I experienced my first trouble with the outfit. It was winter and considerable ice was coming down the river; this choked the intake and offered high resistance to the entering water which caused a greater vacuum in the suction pipe.

When the vacuum reached about 23 inches, the centrifugal pump lost its suction. I stopped the engine and let the pump fill with water from the discharge pipe, and when it was full, I ran the engine up to speed, but the pump would not regain its suction. I then went to the oiler, who had been at the plant for several years, and asked him how he was accustomed to get the suction on the pump after it had once been lost. He replied that it was necessary only to open the valve on the pipe leading from the top of the pump to the sewer, then slow down the engine and run it as slow as possible until most of the air was exhausted from the pump (which would take a couple of minutes), then speed up as quickly as possible and the pump would handle the water all right.

Another time the pump was emptied to pack the stuffing box on the shaft. After this was finished the pump was filled with water and the valve on the pipe leading to the sewer was opened to let out the air while the suction and discharge valves were open. I started the engine and ran it up to speed, but could get no suction. Then it occurred to me that I had not worked the air out before speeding up, so I slowed down the engine and did as the oiler had directed and the pump handled the water immediately. This trouble was experienced only when the vacuum on the suction was over 15 inches.

My theory of this rather peculiar performance is that there was some air trapped in some of the vanes of the impeller when the pump was primed and when the engine was started this air did not get out, unless the engine was run at a very slow speed for a while. The water being heavier than the air, the centrifugal force forced it out of the impeller, leaving the air, which expanded and prevented the pump from raising the water.

K. LAWRENCE.

Kansas City, Mo.

Placing the Blame for Explosions

Every time a fatal boiler explosion occurs in the United States, the papers condemn the laws that cover such subjects and call for laws to be passed that will place the blame where it belongs.

Frequently the men who write these discussions hold up the English law as superior to our own.

In a recent boiler explosion in which the fireman was killed, the coroner's jury came to a verdict of death by negligence, caused by the fireman putting ninety pounds pressure on a boiler that was good for only about forty. In commenting upon this, it was stated that this could not have occurred under the laws of England, for the English Board of Trade would have fixed the blame without fear or favor and the guilty party would have suffered, whether it were the inspector, the boilermaker, the man who owned the boiler, the engineer or the fireman.

But to show that it is about the same everywhere, that the man with the least pull is the man to shoulder the blame, I will relate a case which I observed some years ago.

At the time I was working at the plant of an oil company, about ten miles from Edinburgh, in the capacity of still tender. They had a plant of five Lancashire boilers carrying 150 pounds steam pressure. These boilers were used to supply steam to the stills and to run the pumps. The five were in a battery and connected at the front to a common blowoff pipe (as per sketch) which was located in a trench below the floor, and this trench was covered with an iron plate. In order to blow down the boilers, the fireman removed the plate, put a long-handled box wrench on the cock and, standing with one foot on each side of the trench, opened the blowoff valve.

There were two firemen, and it was their duty to blow the boilers down every shift. They would first open the valve at the far end of the main and then one would take No. 1 boiler and the other No. 2 boiler and in this way they would blow down all of the boilers one at a time.

This they had done a hundred times when one morning the elbow between the main and the blowoff valve on No. 1 boiler burst and one of the firemen was literally boiled alive. Of course, the Board of Trade was prompt in sending what they call in Scotland a "Com-a-tee" to find the cause and place the blame where it belonged. They placed a steam gage on the main between boilers No. 1 and No. 2, closed the valve at the far end of the main and then closed first one and then the other of the whole five blowoff cocks, to prove that the pipe would stand the full boiler pressure. This, however, was after a new elbow had been put in

to replace the broken one. The pipe withstood the shock and this cleared the firm that installed the boilers and piping.

The committee then turned in a report that the fireman was to blame for opening the blowoff too soon. How they arrived at this verdict I cannot see, for upon investigation just after the explosion, both cocks were found closed. Furthermore, they did not look for the burst elbow nor ask any questions along that line. If they had, they would have found that the elbow gave out along an old crack, which, to my knowledge, had been reported a dozen times. But as the pipe was supposed to be under pressure only when blowing down, there was no heed taken of the report. At the inquest the other fireman stated these facts but it was of no use; someone had to be the goat and the dead man was tagged.

PATRICK MULHEARN.

Adams, Mass.

Automatically Operated Oil Pump

During a recent visit to a small power plant I came across an ingenious oiling system which was altogether homemade and seemed to operate perfectly.

A supply tank was attached to the wall near the ceiling, and from this the oil flowed by gravity to the engine. The return from the engine was piped to a filter located in a pit under the floor, and from here the oil filtered into another supply tank. By means of a motor-driven centrifugal pump the oil was forced from the lower tank into the upper supply tank. A float in the latter tank was arranged so as to make or break the contact required to start or stop the motor. The arrangements of this float and automatic switch are clearly shown in the accompanying sketch.

EDWARD T. BINNS.

Philadelphia, Penn.

Kerosene in Boilers

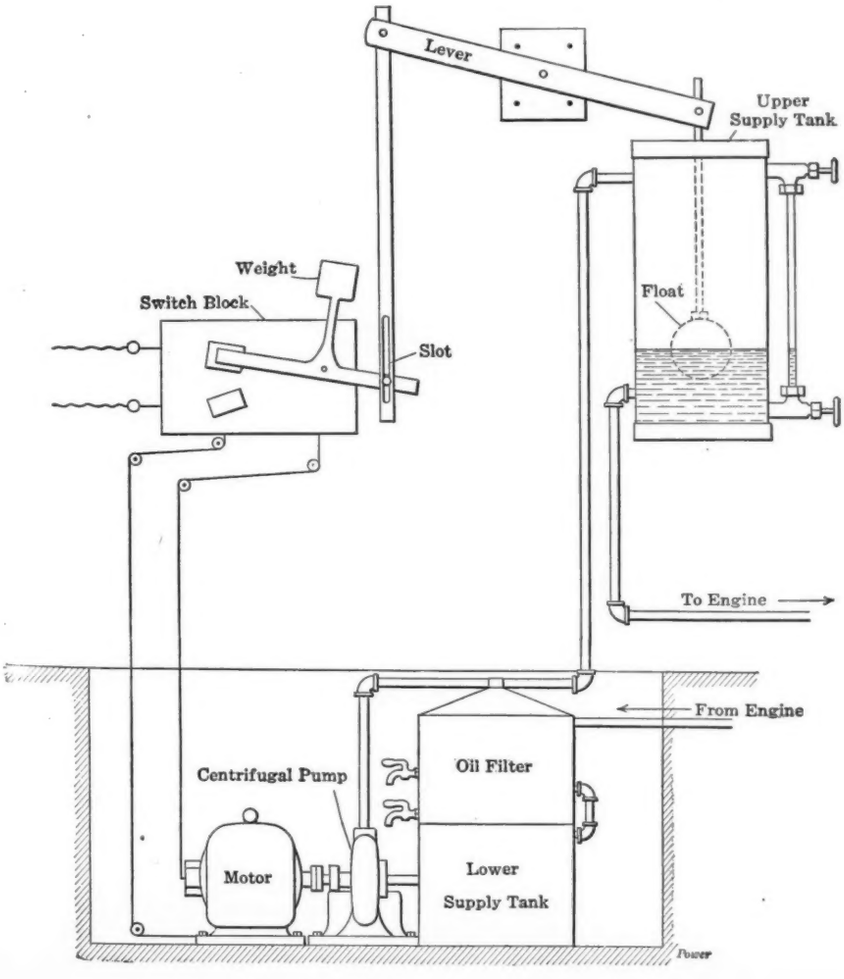
Upon taking charge of my present plant, which is a small Government plant at Riverside school, Okla., I found the heating surfaces of the boilers coated with from ¼ to ½ inch of scale. In fact, I was afraid of their condition, so reduced the steam pressure from 100 to 60 pounds.

I reported the matter to headquarters at Washington, but the only answer which I received was that it was a problem for the engineers. I analyzed the water and talked boiler compounds, but received no reply, so I attached an appliance to the boilers whereby they would receive about one pint of kerosene oil every day. I started this June 1, 1910, and at present the boilers are entirely free from scale.

The superintendent had the engineer from the Fort Hill school come and look over conditions. He contended that kerosene oil should never be applied to boilers. However, I have used kerosene oil in boilers for years with success and the only objection I find is that it takes a little more valve oil. The boilers were tested with cold water at 150 pounds per square inch after they were entirely free from scale and found to be all right. The safety valves were then set back to 100 pounds, which enables me to carry the load successfully.

V. K. STANLEY.

Anadarko, Okla.



DEVICE FOR AUTOMATICALLY CONTROLLING OIL PUMP

Incompressibility of Water

The popular idea that water is incompressible is true in a limited sense only and is probably due to a comparison of this property with that of the gases or other compressible substances. The compressibility of water can readily be shown when comparison is made with such materials as timber, cast iron and steel.

Using the value 0.00005 as the decrease per unit of volume of water due to a pressure of one atmosphere (14.7 pounds) and determining the ratio of this pressure on the unit volume to the deformation it produces, gives

$$\frac{14.7}{0.00005} = 294{,}000$$

This ratio is the modulus of elasticity of the water.

Now the average values of the modulus of elasticity of some of the structural materials, as given by handbooks, are 1,500,000 for timber, 15,000,000 for cast iron, 30,000,000 for steel.

A comparison of these values with that of 294,000 as determined for water shows that the compressibility of water is five times that of timber, fifty times that of cast iron and one hundred times that of steel.

GUY WISE.

Philadelphia, Penn.

Recently at Telluride, Colo., a mountain rat shut down the Ilium station of the power company for a few minutes and threw the city into darkness. The rat had been wandering about the station for some time, causing trouble, and one night, while standing with his hind feet on a steel rafter, he placed his forefeet on a high-voltage wire. Only the inquisitive rodent's skeleton remains to tell the tale.

Have all defects and leaks repaired without delay. It is always best to be on the safe side.

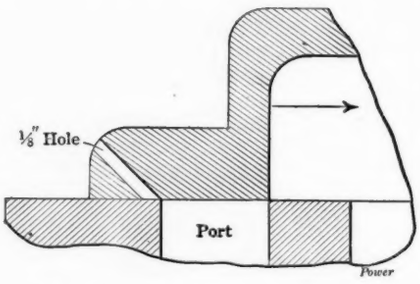
Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Condensing and Noncondensing

Referring to H. W. Benton's article on valve setting for condensing and noncondensing engines, in the September 20 issue, I thoroughly agree with him in that a difference is required, for the purpose of smooth running if for nothing else. However, he does not state the reason for his conclusions.

A number of years ago I was called to a power plant to cure a very annoying knock which occurred at both ends of the engine stroke. The engine was about 500 horsepower in size and running condensing. The knock was not only annoying, but the shocks hammered the crank-pin boxes out of shape and caused heating. A liberal advance of the eccentric cured the noise, but the shape of



LOCATION OF HOLES IN VALVE

the diagrams was changed so much that the engineer suggested that the trouble be remedied in some other way if possible.

After thinking the matter over, I suggested that a number of ⅛-inch holes be drilled in the admission edge of the valve, as shown in the accompanying figure, using care that the holes do not reach far enough on the face of the valve to allow the steam to pass out of the exhaust, before the exhaust edge of the valve covered the port. To make the matter clear, I have shown a section of a plain slide valve with the holes located as stated. We had to experiment for a time to find out just how many holes would be required, but we cured the knock.

Another case was that of an engine fitted with a slide valve with a Meyers cutoff controlled by a right- and left-hand screw, operated by a handwheel. The exhaust piping was arranged with a three-way valve so that a change from noncondensing to condensing could be made without stopping the engine.

When running noncondensing, the engine ran without the slightest jar, but as soon as the three-way valve was thrown over and the engine brought under the influence of the vacuum the knock was terrific.

To cure this case the cylinder clearance was reduced as much as possible with filling pieces, and a new main valve with additional exhaust lap was put on. The main eccentric was advanced as far as possible without interfering too much with the important operation of the valve.

A leaky exhaust valve will often cause a knock by destroying the cushion, while a leaky steam valve will stop it by creating a cushion.

H. S. BROWN.

New York City.

Firing with Gas

Referring to F. P. Wilson's inquiry in the September 27 issue under the title "Firing with Gas," I wish to say that in one of our plants where the boilers are gas fired, the only change made in the furnaces was that of covering the grates with cinders and placing a single row of firebrick midway between the front of the furnace and the bridgewall. The bricks were raised at the ends toward the bridgewall so as to deflect the fire from the burners up toward the shell and over the wall. The furnace fronts were drilled and tapped for 2-inch pipes at points below the fire doors, and pieces of 2-inch pipe, 10 inches long, threaded at one end, screwed in from the inside and pointed toward the center of the firebricks.

The burners were made from two pieces of 2-inch pipe capped at one end and connected to the supply line by a 2-inch tee. These pieces were each drilled and tapped for ¼-inch pipe and ¼-inch nozzles were screwed into them. The burners, when connected up, were placed so that the ¼-inch nozzles just entered the end of the 2-inch pipe in the boiler front. With this arrangement, the gas drew in all of the air needed for complete combustion. The outside gas connections were so arranged that the burners could be turned to one side in case of the failure of the gas and the boilers could be fired with coal without disconnecting any of the burners.

There are mica-capped peek holes in the front and back of the settings so that the condition of the fires can be seen at any time.

The ash and fire doors are kept closed as the burners force in all the air needed with the gas.

J. P. COLTON.

Ohio City, O.

E. P. Wilson's inquiry relative to the above subject in the September 27 issue of POWER made me think that the method used at the plant with which I have been connected for the last three years might be of interest to him.

The accompanying figure illustrates the manner in which the burners are made and connected up. All of the apparatus was made right in the plant. It consists of ordinary pipe fitting. The nozzles are made from ½x3-inch nipples drawn down at one end to about a ¼-inch opening; the other end screws into a ½-inch ell for the upper row of nozzles and a ½-inch tee for the lower row. The mouths of the nozzles extend into the 3½-inch pipes, or air mixer, about 1½ inches. The air mixers are made from old 3½-inch boiler flues sawed into 12-inch lengths and bricked into the furnace-door openings, after the doors had been removed. Six burners equally spaced are placed in each opening, each furnace having three fire doors.

The arrangement and number of burners will, of course, to some extent, depend upon the size and make of boilers to which they are applied. In this case it happens to be 350-horsepower Heine water-tube boilers, arranged two in each battery and two batteries in each division, of which there are two, making a total of eight boilers. Each division is connected to an 80x6-foot diameter concrete stack.

An 8-inch gas main passes in front of each division. At the middle of each furnace a 3-inch pipe rises to the fire door. As illustrated in the sketch, the 3-inch pipe is reduced to 2 inches, branching in opposite directions parallel to the boiler front. Opposite each furnace door three ½-inch pipe taps are made and the burners or nozzles are made up with a ½-inch valve between the 2-inch pipe and the nozzles.

The furnace grates are covered with a course of brick to prevent cold-air currents from entering from below. No checker work has been employed. An ordinary bridgewall as used in the Heine boiler is the only obstruction to the gases except the standard baffle brick on the lower and upper bank of tubes. The gas pressure varies from 4 to 5 pounds

and the fire is regulated by the 3-inch valve in the riser from the main in front of each furnace. The ½-inch valves near the nozzles may also be used for regulating the gas supply. But they are generally used merely when it is necessary to change or clean a nozzle while the furnace is in commission, which seldom happens.

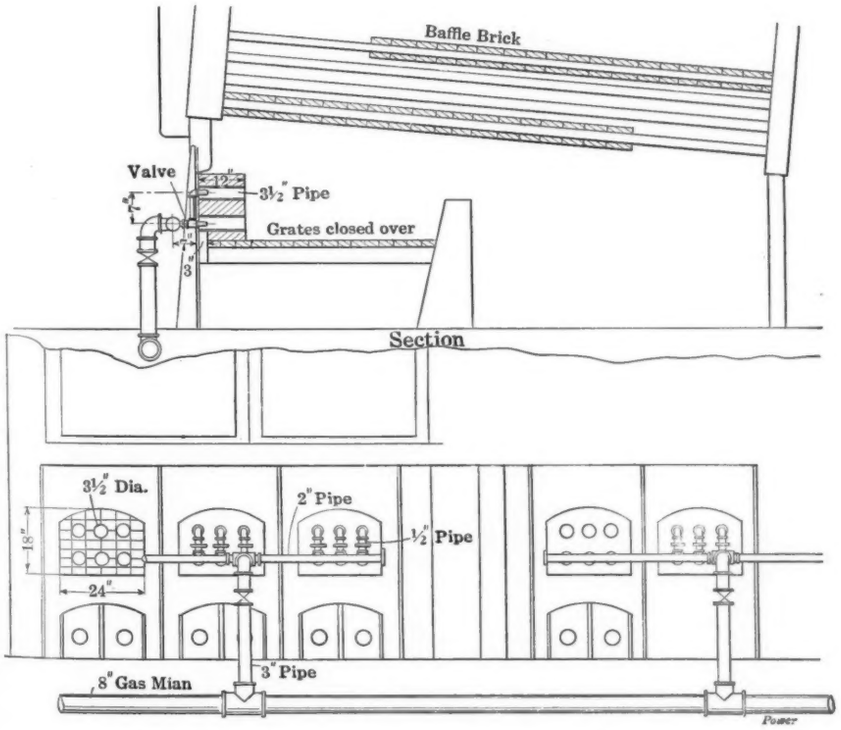
The operation of this system has proved very satisfactory. The boilers have often been overloaded for a day at a time, but there has been no difficulty in keeping up steam and I have yet to see smoke issuing from the boiler stacks.

The important feature in connection with the successful burning of gas consists in having a proper "mix" or air supply at the proper time and place in the furnace. It is, therefore, important that the mouth of the nozzle be placed at a proper distance from the inlet of the air mixer. This distance will depend upon the size of both nozzle and mixer as well as the gas pressure, although the latter will tolerate a comparatively large range without affecting the mix to any great extent. When banking fires, a piece of sheet iron, cut round and a little greater in diameter than that of the mixer, with a slot large enough to fit over the nozzle, is placed over the opening of each mixer and the gas turned down to a suitable degree.

It is not easy to say what causes the difficulties Mr. Wilson states, as he does not go into the details of the conditions under which he has been burning gas under his boilers. But it is likely that the trouble is due to either improper "mix," lack of draft or admitting cold air into the furnace at other places than where the gas enters and ignites.

T. C. THOMPSON.

Chanute, Kan.



GENERAL ARRANGEMENT OF FURNACE AND BURNERS

Leakage through a Piston Valve

A careful reading of the article under the above title in the October 11 issue shows that its author spent a great deal of labor to *prove* that he was working with a leaky valve. He then fell into the fatal error of making a sweeping generalization from a single case. He finds this piston valve leaky; therefore, all piston valves are leaky.

And this is his process: In a small engine, he measures up the bore of the valve chest and the diameter of the valve and orders from the engine maker a special valve for experimental purposes. He may get a good-fitting valve or a bad-fitting valve. Upon test, he finds it leaks badly; therefore, he concludes, all piston valves leak badly. He expresses his conclusions in the following language:

"A significant fact, although one often lost sight of, is the enormous proportions assumed by the leakage past the piston valve of a steam engine; it reaching, in small engines, as high as 22 per cent. of the total steam consumption. When one stops to consider that this represents energy lost, its importance will at once be comprehended. The method herein described was devised by the author to measure this leakage under actual running conditions."

As to his "method" of measuring the leakage, he does not do it by direct measurement. On the contrary, a series of experiments is carried out, from which he undertakes to deduce certain laws governing leakage; and then, by a number of more or less elaborate calculations based on various assumptions, he attempts to arrive at the "actual" leakage.

His plan was to use a special valve which controlled the steam in the usual manner for the head end only, while the crank end of the valve was merely a blind plug traveling in the valve bushing. The engine operated as a single-acting engine, and the crank end of the cylinder was connected to a condenser to draw off and measure whatever steam leaked past the piston and the plugged crank end of the special valve.

He describes his calculations as follows:

"In making the calculations the assumption was made that the leakage was proportional to the pressure.

"The total leakage per hour through the various lengths of the crank end of the special valve, together with that of the piston, was reduced to equivalent leakage at 100 pounds pressure, and then the total leakage per hour at 100 pounds pressure was determined for the piston. The piston leakage being subtracted from the total leakage for each of the various lengths of the special valve through which leakage occurred, gave the net leakage through each of these various lengths, per hour at 100 pounds pressure. These results are shown by the curve, Fig. 5. Next, the equivalent distance constantly leaked through when the crank end of the special valve was at each of the various lengths was calculated, the results being shown by the curve in Fig. 6.

"The actual leakage per hour at 100 pounds pressure through the standard valve was now calculated. With the engine running normally, leakage occurred as follows:

"1. Between the steam chest and the exhaust passage, through the spaces between the steam ports.

"2. Through the spaces equal in width to the steam ports between admission and cutoff.

"3. Between the cylinder and the exhaust passage, through spaces equal in width to the steam ports between cutoff and release and between compression and admission.

"In order to determine the amount of these leaks, it was necessary to know the equivalent distances constantly leaked through and the pressure at which the leakage took place. The valve travel at each of the various horsepowers was determined, and knowing the valve travel, the equivalent distance constantly leaked through in each case, for each of the various horsepowers, was calculated.

"Leakage 1 occurred at a pressure

equal to that of the steam chest, as shown by the steam gage. Leakage 2 was due to this same pressure. Leakage 3, between cutoff and release, occurred at a pressure due to the average mean effective pressure of the expansion curves of the indicator diagrams for that particular run, and between compression and admission, occurred at a pressure due to the average mean effective pressures of the compression curves of the indicator diagrams for that particular run. Each of the above leaks was determined per hour at 100 pounds pressure by use of an equivalent distance curve.

"This same method was used for each of the various horsepowers from 5 to 15, and the results were plotted as shown by the curve, Fig. 6.

"The ratio of the total leakage through the standard valve to the total steam consumption at the various horsepowers was next determined, the results being shown by the curve in Fig. 7."

Thus it is seen how far he has strayed afield in the realm of supposition, hypothesis and imagination, away from ascertained or proved facts. Now all such roundabout and laborious "methods" are quite unnecessary. If a piston valve, or any other kind of valve, is leaking excessively, it is immediately and directly shown in the steam consumption of the engine. And that this particular engine was wasteful would be shown directly by the economy test which the author first made but does not report in his article. He says, however, that when the engine was developing 15 brake horsepower, the leakage of steam through the valve was 200 pounds (per hour, presumably), and that this leakage amounts to 22 per cent. of the total steam used. The total steam used would, therefore, be 200 pounds per hour divided by 0.22, which is about 910 pounds per hour, or about 60 pounds per brake horsepower per hour.

None of his elaborate calculations is needed to prove that this engine did indeed have a leaky valve. But what has this to do with other piston-valve engines which are commonly showing a steam consumption of about half what he found?

He calculates that the valve leakage of this engine was 22 per cent., while the total steam consumption was about 60 pounds per brake horsepower per hour. Now, if this valve leakage of 22 per cent. was stopped entirely, the steam consumption would then be 78 per cent. of 60 pounds, or about 47 pounds per brake horsepower per hour. What, then, would be the valve leakage of piston-valve engines which commonly show an economy of 35 pounds per brake horsepower per hour?

About three years ago, we made a careful investigation of piston-valve leakage, by running a series of economy tests on a single engine of average size, using in succession twenty different valves. All conditions were kept unchanged, except that a different valve was used for each test. Fairly high-pressure steam (150 pounds at the throttle) was used to develop the leakage to the full extent. Fifteen valves were made to have the ordinary commercial fit, and by direct measurement the steam consumption at full load was found to range from 26 pounds, the minimum, to 27 pounds, the maximum, per indicated horsepower per hour.

Then, a series of tests was made with valves intentionally made loose and of bad fits, to develop the loss thereby resulting. A valve two to three thousandths of an inch under size is plainly a poor fit. When moving it by hand it feels loose—any engineer would observe it. Three such valves raised the steam rate to 27.05, 27.4 and 27.8 pounds, respectively.

Two valves were then made ten thousandths of an inch under size. This is a fit so bad that no engineer would tolerate it—the valve could be rattled by hand in the valve chest. With these valves the engine tested out a steam rate of 32.1 and 32.7 pounds, respectively. This remarkable showing is explained by the long surfaces of the valve and the distance the steam had to travel through the fit in order to leak.

A. L. Ide & Sons.

Springfield, Ill.

License Laws

I am an engineer because I like engineering better than any other class of work and because, like any other human being, I must live. As there must be engineers as long as there are engines, I naturally picked out the trade I liked best, which I believe is one of the main essentials to success.

I believe in a strict license law which will allow only competent men to have charge of power plants; also, in a strict boiler-inspection law, which is of great importance to the engineer as well as the owner. Furthermore, I think that it is well worth while for the engineers to demonstrate to the owner that he is worth a better salary if the one he is getting is too small. After proving his ability an engineer should not be afraid to ask for an increase. If the owner will not pay what the work is worth, look for a man who will. However, do not leave one job until you see a chance to do better; it is of no use to cut off the nose to spite the face.

I am not posing as a "public benefactor" in demanding better laws, as a writer in a recent number of Power called progressive engineers. It has been said by some engineers that license laws are merely an attempt to limit the number of engineers for the purpose of keeping wages up by creating a demand in excess of the supply. Assuming that this is true, it is no more than is being done in any number of other classes of work. If you go into a machine shop you will find that the old machinists have a standing grudge against the apprentice. The same may be said of all the other trades and professions, and the engineering business in all its branches is of no small magnitude.

Some writers state that the engineer receives less wages than men employed in other trades and also that the wages in license-law States are no higher than in States where there are no license laws. I suggest that Power collect data on the wages, hours and general conditions of engineers in different States and on the size of plants, etc., and print the results in a tabular form similar to that used some time ago in regard to cylinder lubrication.

In this way some comparison could be made as to the benefit of license laws. It seems to me that some such data could be collected which would give us some definite information concerning the conditions of the engineer.

J. Case.

Hyattsville, Md.

G. W. Johnson asks in the October 11 issue: "In a no-license State, how shall the employer decide upon the merit of an applicant who is unknown to him unless he judge by letters of recommendation which too often are worthless?"

Again, he says:

"Take the employer who believes, on principle, in having everything of the best—equipment, supplies and operatives, and is willing to pay the price. . . . Please state, you nonlicense men, how you, in his place, would judge the fitness of applicants?"

It is the business of the Government to *protect* the people in their peaceful pursuits, their homes, etc. But the Government exceeds its power when it undertakes to guarantee to employers efficient workmen. It is within its rights when it prevents the employment of a person in a position with which that person is unfamiliar, thereby becoming a menace to the public. The Government has not the power to weed out the inefficient; it has the power to weed out the menaces.

Again, I quote:

"Recently one or two engineers have expressed the opinion that license laws would have but little beneficial influence. . . . I should like to call their attention to the fact that with the enforcement of a proper law, an immense number of operating misfits would at once be obliged to leave the field clear for those who by virtue of a superior intelligence and ambition already possess the necessary qualifications."

I cite this passage in support of my statement in the September 27 issue, that the motive of the engineers behind this license-law movement is to make scarce

the supply of engineers that the scarcity may fructify to those remaining.

The reason why license laws will not accomplish the weeding out of the inefficient is because it is against the law of evolution. According to this law, man collectively will improve at a given rate; this rate is regulated by two factors which are equally important, conditions and the man. Conditions present the necessity and man, individually, will or will not rise to the occasion. If he lacks the temperament, the ambition, the energy, the intelligence, the physique or any of the requirements that go to make up the human factor he cannot meet the requirements. If the requirements call for a standard which cannot be met by a sufficient number of capables, then the standard must yield and the necessity will be met by that degree of efficiency held by those who cannot attain the standard.

If there are a sufficient number of men whose knowledge and experience qualify them to *safely* operate a steam plant so that it is not a menace to the public, a license law would have the effect of putting to work those who are found to be *safe* as long as they are to be had, to the exclusion of the unsafe.

Attention is called to the emphasis on *safe* because any license law or examining board that ventures beyond the point where the public is interested and proceeds to examine into a man's *ability* or *efficiency* is working in the interest of the employer and not in the interest of the State for which it was created and is, therefore, illegal.

It is my opinion that every engineers' examining board in the country is transgressing the law in the interest of the employer and to the hardship of the engineer.

J. J. NIGH.

New Haven, Conn.

Some engineers seem to feel that it is somewhat of a hardship to have to go to the expense of paying a license fee. This may be true, but the examination will be worth the expense as a review. Many a member of an examining board could not answer nearly as many questions as some engineers. On the other hand, some very able engineers could not stand an examination, although they can do things and do them in the best possible manner, and are safe to intrust with the care of a steam plant. And, again, there are exceptions. I once met a man who was an engineer, had his license, had been examined by one of the boards, yet with all of this the man could not correct a faulty valve gear. This was not the fault of the license-law system.

The license law will cut out, to a certain extent, the opportunity for a man to say that he is an engineer when in reality he is not. You would not like to sit in a passenger coach behind an engineer that you knew nothing about unless you knew that the railroad had a man in the cab that had stood a thorough test. The railroads go so far as to have special cars in which to conduct examinations and secure practical illustrations of a man's ability. Why could not the stationary engineer be tested under similar conditions?

The United States Government has found that in order to have men to fill the places safely it is necessary to have examinations. They do not take the man on what he says but on what he can show by actual demonstration. Steam machinery is becoming more complicated each day and higher pressures are coming into use. Hence, plants should not be intrusted to those who we do not know are all right. In times gone by we may have gotten on fairly well without license laws but modern conditions necessitate suitable regulating laws. I see no reason why the engineer should complain. It will certainly better his condition. In some cases he may have to read up some but there is plenty of time for this, and those who are prepared will not object to being examined.

C. R. MCGAHEY.

Lynchburg, Va.

Installing Globe Valves

The globe valve that controls the steam to the boiler feed pump is frequently adjusted by the attendant and is never wide open and seldom closed. For this reason it will always work better with the pressure under the disk, for with the lower pressure on the packing the valve can be closed and packed at will, and if the disk comes off the stem the pump is not thrown out of service.

A globe valve should never be put in the feed line with the pressure from the boiler under the disk as the pressure from the pump will then be on the top and if the disk should come off the stem no water can be put into that boiler until the valve has been repaired. Nor can the valve be packed while the feed pump is running.

The blowoff valve on the water column should have the pressure from the boiler on the top of the disk for the reason that it is often used by inexperienced firemen and left loose enough to leak and so destroy the seat.

A globe valve should never be used on the steam line between the boiler and the main header. If the valve is put on with the pressure from the line on the top of the disk it cannot be packed while steam is on the line. This would be bad, for steam may have to be kept on the line for months at a time. On the other hand, to install the valve with the pressure from the line under the disk would be even worse, for no sane engineer or boiler inspector would enter the boiler with high-pressure steam on the line, because often the valve yoke breaks and the stem threads strip, in which event it would not be hard to guess what would happen to the man on the inside.

When I took charge here the angle blowoff valves were attached with the pressure under the disks, a Homestead valve being between each angle-valve and boiler. When the angle valve leaked, which happened about every three days, the Homestead was closed, a new lining put in the iron disk of the angle valve and the seat turned off. When I installed new blowoff piping I took out the Homesteads and changed the angle valves so that the pressure came on the top of the disks, with the result that the valves now give very little trouble.

JAMES JOHNSON.

Hackett, Penn.

Boiler Feeding by Centrifugal Pumps

The editorial in the issue of October 11 under the above heading is worthy of discussion. There are many points in favor of the use of the centrifugal pump as a boiler feeder, both motor and turbine driven. I think that in either case the speed should be variable in order to meet sudden demands. In a plant with a steady load the feed water can be fed at a constant rate, but in street-railway or hydraulic-elevator plants the water must be supplied in accordance with the demand for power, the demand varying with the number of cars running and the load they carry. Another important point in favor of the centrifugal pump is found when an open heater is used, and the exhaust steam from the pump is used to heat the water in the heater. Reciprocating pumps usually require considerable cylinder oil which is exhausted into the heater and is constantly getting into the boilers through poorly working oil separators. The centrifugal pump should be greatly appreciated for this one reason.

The first cost, however, is slightly against them, but this may possibly be reduced as the demand increases.

I have seen many centrifugal pumps used in hot-water heating systems to good advantage under low pressures and handling water at 190 degrees Fahrenheit and above. To equip the centrifugal pump with a turbine engine makes it independent as a steam pump and requires much less care in operation.

Pressures are now increasing very rapidly and most new installations consist of boilers with a working pressure of from 175 to 200 pounds. When centrifugal pumps are produced that will deliver water of from 200 to 210 degrees against these pressures at a variable speed without the possibility of the water hammering or the pump becoming steam-bound, the demand will increase rapidly.

R. A. CULTRA.

Cambridge, Mass.

Comparison of Lubricating Greases

By Horace W. Gillett

In a series of comparative tests upon corresponding samples of mineral and tallow greases, in which both Olsen and Thurston testing machines were used with varying bearing pressures, the tallow greases showed a lower melting point and a lower coefficient of friction.

Most engineers and users of lubricants agree that there are certain bearings on which, for the sake of convenience, cleanliness or economy, it is advisable to use a lubricating grease. But having gone that far, they consider that "grease is grease," and that one kind is as good as another.

Commercial greases differ in chemical composition vastly more than do lubricating oils, and as it was almost certain that their lubricating properties differed also, it seemed worth while to the writer to make a series of actual friction tests of the common greases on different oil-testing machines, particularly as the literature of lubricants show such a woeful lack of impartial investigations.

Before taking up the actual results of the work, it is advisable to consider the types of greases now in use, in order to see what a knowledge of their composition and properties would lead one to expect of them as efficient lubricants. Gear greases, graphite and mica greases will be left out and the discussion confined to those greases commercially used and widely advertised for service where a high-grade and efficient lubricant is demanded.

There are only two types of greases that command the serious attention of the engineer, the mill superintendent or the automobile owner; these are the so called mineral greases and the tallow greases. The mineral greases consist of mineral oils, thickened with a percentage of lime soap, varying according to the consistency desired. These have a high melting point, usually from 150 to 200 degrees Fahrenheit. The tallow type of greases consists mainly of some hard animal or vegetable fat such as tallow, with a small amount of mineral oil and a small amount of soap, usually a potash soap. As a rule, they melt at from 75 to 125 degrees Fahrenheit, although at room temperature they are much harder than the corresponding mineral greases.

Any grease would be a hindrance to the bearing rather than a help, if, when in action, it were in the same condition in which it exists in the grease cup. In its comparatively hard and sticky consistency it offers resistance to the moving parts; hence before it begins to lubricate, it must melt. In other words, the grease should become an oil when it comes in contact with the bearing. There is no doubt that the actual contact surfaces of a grease-lubricated bearing are at as high a temperature as the melting point of the grease, although the whole bearing and journal seldom reach that temperature, because of radiation to the air and conduction to other parts of the machine.

When starting, the shaft generates heat by friction until the grease melts and becomes a true lubricant, at which point it begins to cut down the friction. To keep this initial friction and consequent power loss small, one should use a grease of as low a melting point as is compatible with having the grease stay in the cup without waste. This judgment is used by the engineer who uses a summer and a winter grade of grease, colder weather allowing him to use a grease of a lower melting point, or softer consistency. But the melting point is not the only factor; the friction-reducing power of the oil that the melted grease forms must be considered.

TABLE 1. RESULTS WITH THURSTON MACHINE.

Type.	Melting Point, Degrees Fahrenheit	Percentage of Soap.	Kind of Soap	Percentage of Free Acid as Oleic.	Average Coefficient of Friction.	Final Coefficient of Friction after 3-hour Run	Maximum Temperature of Bearing above that of room, Degrees Fahrenheit.	Final Temperature of Bearing above that of room at end of 3-hour run, Degrees Fahrenheit.
A, mineral, summer	167	38	Lime	Trace	0.075	0.75	70	68
B, mineral, summer	178	20	Lime	0.3	0.054	0.050	70	58
C, mineral, winter	165	23	Lime	6.1	0.063	0.063	76	65
D, mineral, winter	163	16	Lime	0.0	0.057	0.054	69	58
E, mineral, winter	142	19	Lime	Trace	0.046	0.046	58	50
F, tallow No. 3, winter	125	1.4	Potash	0.0	0.022	0.012	38	18
G, tallow No. XX, summer	120	2.1	Potash	0.0	0.029	0.018	45	32
H, lard oil	41	0.0	.....	.....	0.011	0.010	13	12

TABLE 2. RESULTS WITH OLSEN MACHINE.

	500 Lb. Total Pressure on Bearing = 86.5 Lb. per Sq.In.			1000 Lb. Total Pressure on Bearing = 173 Lb. per Sq.In.			1500 Lb. Total Pressure on Bearing = 253 Lb. per Sq.In.			2000 Lb. Total Pressure on Bearing = 345 Lb. per Sq.In.			2500 Lb. Total Pressure on Bearing = 431 Lb. per Sq.In.			Average Final Coefficient of Friction.	
	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	All Pressures.	All Pressures and Grades.
SOFT GREASES																	
Mineral, X	28	0.0075	0.0062	46	0.0111	0.0085	68	0.0120	0.0103	86	0.0138	0.0125	83	0.0196	0.0152	0.0106	.....
Tallow, Albany No. 1	22	0.0075	0.0050	31	0.0084	0.0075	44	0.0113	0.0093	67	0.0135	0.0123	75	0.0144	0.0117	0.0092	.....
MEDIUM GREASES																	
Mineral, Y	24	0.0126	0.0080	39	0.0114	0.0095	50	0.0123	0.0107	65	0.0140	0.0115	75	0.0140	0.0120	0.0102	.....
Tallow, Albany No. 3	23	0.0088	0.0056	36	0.0100	0.0080	50	0.0112	0.0103	64	0.0140	0.0115	75	0.0140	0.0116	0.0094	.....
HARD GREASES																	
Mineral, Z	28	0.0113	0.0090	55	0.0132	0.0108	72	0.0136	0.0113	76	0.0141	0.0117	87	0.0132	0.0116	0.0109	.....
Tallow, Albany XXX.	26	0.0090	0.0067	45	0.0131	0.0100	49	0.0124	0.0100	67	0.0127	0.0100	69	0.0121	0.0100	0.0093	.....
Mineral	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.0106
Tallow	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.0093

To find out which is actually the best type of commercial grease, two series of friction tests were made, one on a small Thurston testing machine and the other on a large Olsen machine. The series of tests made upon the Thurston machine will be taken up first. This machine was run throughout the series of tests at 320

revolutions per minute and with a 60-pound pressure on the bearing, which is equivalent to 240 pounds per square inch upon the projected area. The bearing was thoroughly cleaned, smeared with the grease to be tested, the cup cleaned and filled, and the machine then run for three hours, the coefficient of friction and the temperature of the bearing being read every five minutes.

Seven kinds of commercial greases were thus tested. These consisted of five samples of the mineral type; three

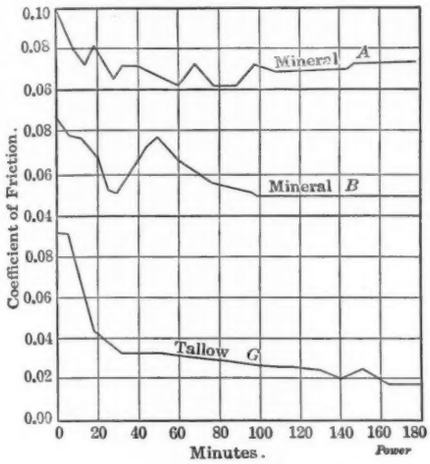


FIG. 1. SUMMER GREASES

of the winter grade and two of the summer grade, made by three different representative firms; and two samples of tallow grease—Albany No. 3 and No. X X. Besides these greases a pure lard oil was used as a standard of reference, so that tests made on other machines or by other observers might be compared with our results.

In order to compare summer and winter greases with others of their own class, they were divided into two sets, those called *A*, *B* and *C* representing the summer grade, and *C*, *D*, *E*, *F*, the winter grade. The lard oil is designated by *H*. It is perhaps straining a point to put Albany No. 3 or *F* in the winter class, as it is quite a stiff grease and several softer grades are made, but for this series of tests none of those softer grades were available.

The results of this series of tests are given in Table 1.

The higher melting point and higher soap content of the mineral type are shown by the table and it is also of interest to note that although the tallow greases migh be expected to become more or less rancid, as they are made largely from natural fats, this is not the case, the mineral greases being the ones that show free acid from the fats used in making the soap content. If palm oil is used in making the stiffening soap, as is often the case because of its low price, any of that oil which is not neutralized will sooner or later give rise to free acid. Although the 6 per cent. of acid found in one of these mineral greases is higher than that type will usually run, a trace of free acid is very often present in mineral greases. None of the tallow greases that the writer has examined have shown any free acid.

The coefficient of friction curves plotted against time is given in Figs. 1, 2 and 3, the first representing the summer greases, the second the winter greases and the third the lard oil.

Without a single exception the tallow greases, with their low melting point and their low soap content, gave better results than the other types—results not much inferior to lard oil itself. To check the conclusions derived from the tests made with the Thurston machine and to study the effect of varying the pressure, a series of tests was made on an Olsen oil- and grease-testing machine belonging to Sibley College of Mechanical Engineering, Cornell University.

The pressure used on the Thurston machine was 240 pounds per square inch, while that on the Olsen could be raised to 431 pounds per square inch, or 2500 pounds actual pressure on the test bearing, which had an area of 5.8 square inches. The diameter of the journal on the Olsen machine was 3¾ inches and its length 3½ inches. The journals of both machines were of steel with the bearing of the Thurston of brass, and that of the Olsen of babbitt. The latter was run at from 215 to 240 revolutions per minute, which was equivalent to a speed of from 212 to 235 feet per min-

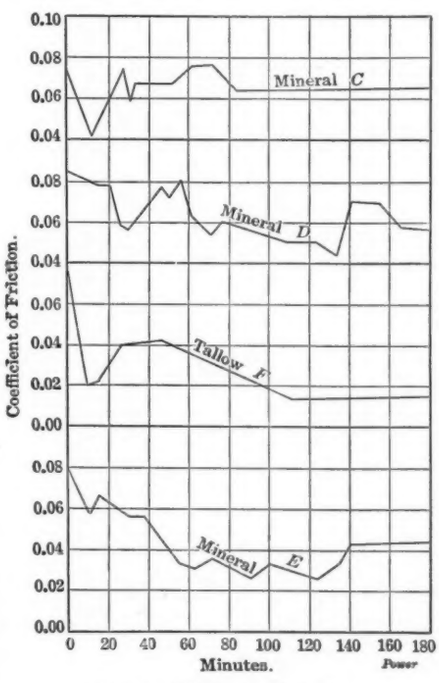


FIG. 2. WINTER GREASES

ute. Runs were made at various pressures, as shown in Table 2, and the speed, friction and temperatures were read every ten minutes. Each run lasted one hour, the temperature and coefficient of friction in every case becoming constant before this period was over. The greases were fed at such a rate as would maintain in each case the lowest coefficient of friction possible with the grease used.

As there was not enough remaining of the samples used on the Thurston machine to make all the tests desired on the Olsen, the opportunity was seized to make a closer comparison between the tallow greases and the mineral greases of corresponding grades. To this end three grades of Albany grease, hard, medium and soft, were selected and matched by three samples of mineral greases, hard, medium and soft, which were recommended as suitable for the same classes of work for which these grades of Albany grease are recommended. Of course, the consistency of the tallow greases at room temperature was harder than that of the corresponding mineral grease, owing to their different composition. To compare with Albany No. 1, which is a soft grease,

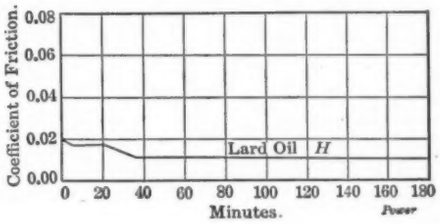


FIG. 3. LARD OIL

a mineral grease was selected which for convenience is designated in Table 2 as *X;* to compare with Albany No. 3, which is a medium grease, a mineral grease *Y* was chosen; and for Albany XXX, which is a hard grease, a mineral grease *Z* was selected.

The results of the tests are given in Table 2, and in Figs. 4, 5 and 6 are shown graphically the final coefficients of friction, obtained at the end of the hour's run, for each pressure.

The mineral grease contained so much soap that the residue taken off the bearing after a run consisted of oil and hard particles of soap; that is, the oil had melted away from the soap, and the grease was no longer homogeneous. The grease parts with the oil first and what is finally left in the cup is practically nothing but soap. This will not show up in an hour's test, of course, but where the grease is used long enough for the oil to be exhausted, as is the case in practice, this is a vital point.

In the table are given both the average coefficient of friction during the whole run and the final coefficient. The latter is the more important, as it shows a condition in which the bearing will run all day long. But if a grease that gave a low final coefficient of friction gave a very high one on starting, then the average coefficient of friction would give valuable information as to the value of the particular grease on a bearing that was to be run only intermittently. Whether the average or final values of the coefficient of friction are considered

or whether the temperature of the bearing is taken, at all pressures the tallow greases showed a distinctly greater friction-reducing power than did the greases of the mineral-oil type.

It will also be noted that the stiffer

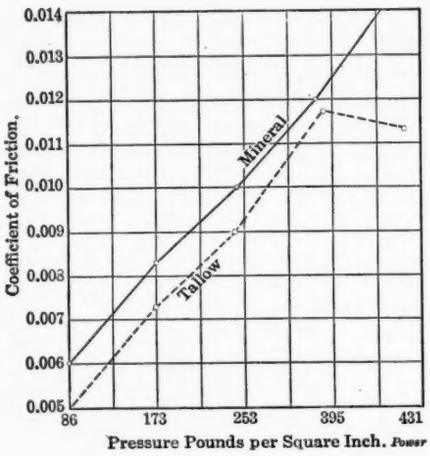


FIG. 4. SOFT GREASES

greases gave better results at high pressures than the softer grades, Albany XXX, the hardest grade of tallow grease, giving equally as good results at 431 pounds per square inch as at 173 pounds, while the softer greases showed an increase in friction with an increase of pressure. Hence, to get the best results not only must the type of grease be determined but also the grade best adapted to the particular problem under consideration.

The difference between the tallow and mineral types, or between the hard and soft mineral greases, does not show up as great on the Olsen machine as on the Thurston, even at practically the same pressures; the value obtained for the coefficient of friction on the former is less than that obtained on the latter. This throws considerable light on the problem, and upon examining the two bearings the reason is found. The journal and bearing on the Olsen machine, which were new and well kept, were as smooth as it is practicable to make a bearing, while the bearing of the Thurston machine was not very smooth, as it had not been used for several years before this series of tests. It was ground down as smooth as possible, but was still rather rough. In other words, on the Thurston machine the grease had to do more lubricating, and

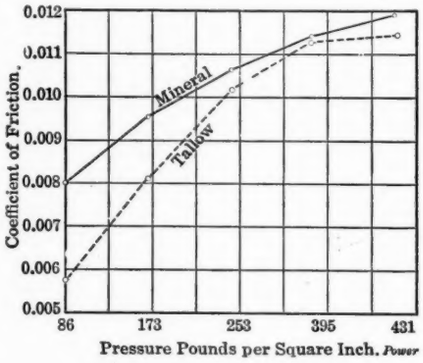


FIG. 5. MEDIUM GREASES

in such a case the better lubricants would show an increased efficiency, as they have a wider field to work in. This shows not only the superior ability of the tallow grease to lubricate under trying conditions, but also that in making tests to show up differences in the lubricating power of oils or greases, the test bearing should not be in too good a condition. Even on the good bearing, the average decrease in friction in the whole fifteen sets of runs obtained solely by substituting a tallow grease for a mineral one is about 12 per cent.

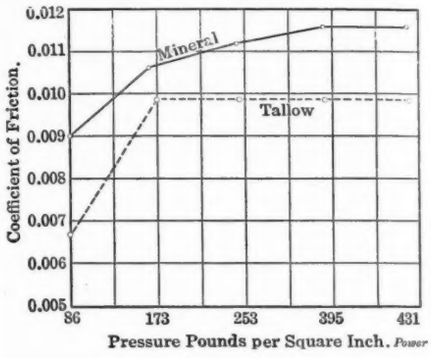


FIG. 6. HARD GREASES

To sum up the results of the test it may be said that greases with a low melting point and with a low soap content should be the most efficient theoretically; the tallow greases have a lower melting point and a lower soap content than mineral greases; in actual tests on two oil-testing machines, the tallow greases consistently showed their superiority to the greases compounded from mineral oil and lime soaps, not only at one pressure on the bearing, but at all pressures.

Low Pressure Steam Turbines*

By H. G. Stott

In the Interborough plant the installation of low-pressure turbines more than doubled the economic capacity of the plant and resulted in the remarkable unit thermal efficiency of 22 per cent.

Several installations of low-pressure turbines have been made during the last year with satisfactory results. The largest

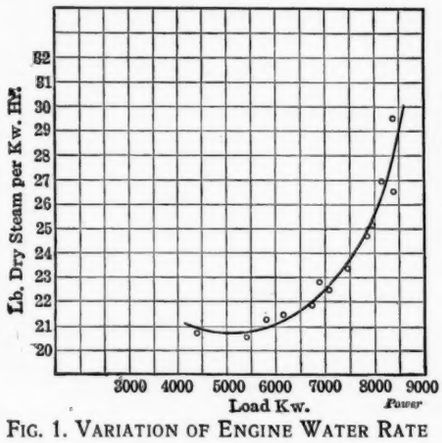


FIG. 1. VARIATION OF ENGINE WATER RATE WITH LOAD

installation has been that of the Interborough Rapid Transit Company in New York, which has installed three 7500-kilowatt units and is now engaged in installing two more units of the same size.

Brief extracts from a very elaborate series of tests on these low-pressure units, which have already been published, follow, as they may be looked upon as typical of what may be expected in other installations under similar circumstances.

Each of the 7500-kilowatt low-pressure turbine units receives its steam from a Manhattan type of Corliss engine having two 42-inch horizontal high-pressure cylinders and two 86-inch vertical low-pressure cylinders. The high- and low-pressure pistons on each half of the unit connect onto the same crank, so that the unit consists of two compound engines with the generator between them running at 75 revolutions per minute and with a 60-inch stroke. The revolving field of the generator is built up from riveted steel plates of sufficient weight to act as a flywheel for the engines on each side of it.

The high-pressure valves are of the poppet type and the low-pressure valves of the Corliss type. These units were rated at 5000 kilowatts with 50 per cent. overload capacity, giving their most economical results between 4000 and 5000 kilowatts. As far as the heating limit was concerned it has never been reached as the engine becomes too uneconomical to justify operation before the maximum capacity of the generator is reached.

Fig. 1 gives the engine water rate when running noncondensing, and Fig. 2 gives the water rate for the low-pressure turbine, with the steam pressure entering it varying according to the load on the engine from one pound above the atmosphere to five pounds below, with a cor-

*Portion of the report of the Committee on Power Generation of the American Street and Interurban Railway Engineering Association, Atlantic City, October 10-14.

responding condenser vacuum of 28.5 inches when referred to a standard barometer of 29.92 inches.

The opposite slopes of Figs. 1 and 2 result in the combined water rate shown in Fig. 3, giving an almost flat water rate for the combined unit between the wide limits of 6500 and 15,500 kilowatts.

To show the gain made by the installation of the low-pressure turbines the water rate of two engines is given at *B*, Fig. 3, for comparison, showing an average improvement in the combination unit, curve *A*, of over four pounds per kilowatt-hour.

Fig. 4 shows water rate guaranteed for a typical modern 12,000-kilowatt standard high-pressure turbine unit when running under the same steam pressure and vacuum as obtained for the combined unit, as shown in Fig. 3.

All these curves shown are results obtained with saturated steam only.

Fig. 5 shows the heat distribution between the various parts of the unit, showing the remarkable unit thermal efficiency of over 22 per cent.

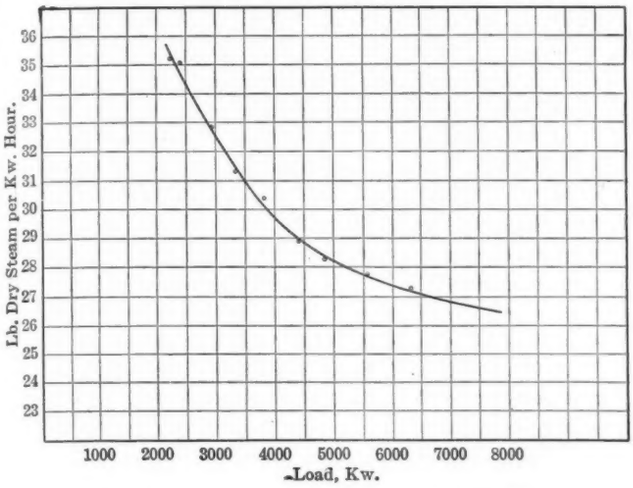


FIG. 2. VARIATION OF TURBINE WATER RATE WITH LOAD

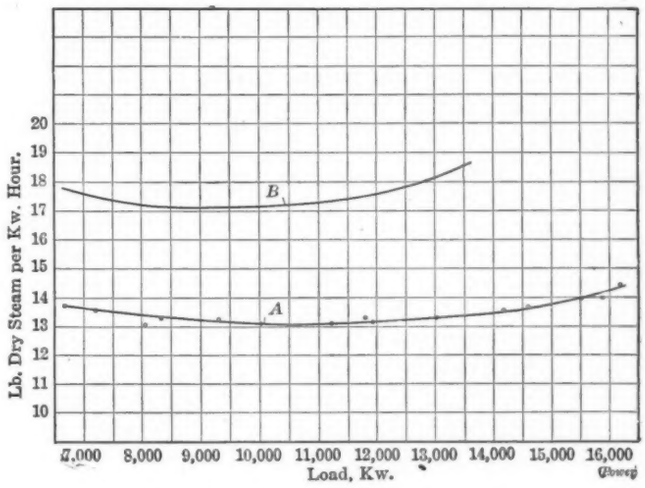


FIG. 3. VARIATION OF UNIT WATER RATE WITH LOAD

The results obtained by the installation of low-pressure turbines are as follows:

(*a*) An increase of 100 per cent. in maximum capacity of plant.

(*b*) An increase of 146 per cent. in economic capacity of plant.

(*c*) A saving of approximately 85 per cent. of the condensed steam for return to the boilers.

(*d*) An average improvement in economy of 13 per cent. over the best high-pressure turbine results.

(*e*) An average improvement in economy of 25 per cent. (between the limits of 7000 and 15,000 kilowatts) over the results obtained by the engine units alone.

(*f*) An average unit thermal efficiency between the limits of 6500 and 15,500 kilowatts of 20.6 per cent.

One of the novel parts of this installation is the use of the induction type of generator on the low-pressure turbines. The reason for adopting this type of machine was that with the enormous short-circuit capacity of the system, amounting to over 300,000 kilowatts, the addition of large synchronous turbo-generators would result in a largely increased short-circuit capacity and it was feared that the switching apparatus would soon reach its limit of safe operation. The addition of induction generators, on the other hand, adds nothing to the short-circuit capacity and tends to help the regulation by adding so much stored energy or flywheel capacity. The results obtained have fully justified the installation and the simplicity of operation is most marked.

In general the installation of low-pressure turbines may be recommended wherever there are good engines already installed or in the case of a new installation where the load factor and the coal cost are high. In plants having a large installation of a good type of reciprocating engine, the low-pressure turbine may be added at a total cost, including new condenser, auxiliaries, foundations, piping, etc., of not to exceed $25 per kilowatt, thus bringing down the average overall investment per kilowatt of the entire plant and so reducing the fixed charges per kilowatt-hour.

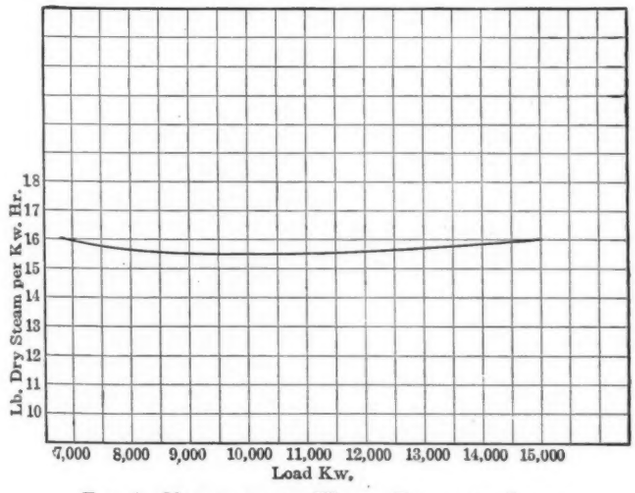


FIG. 4. VARIATION OF WATER RATE AND LOAD, HIGH-PRESSURE TURBINE

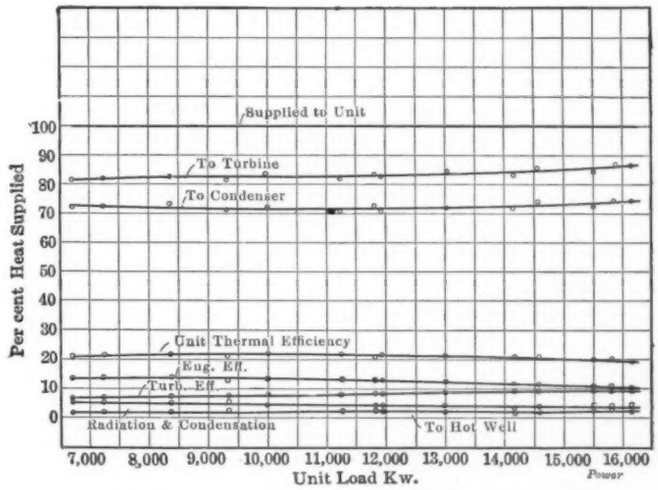


FIG. 5. VARIATION OF HEAT DISTRIBUTION WITH UNIT LOAD

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Split Blowoff Pipe

If the blowoff pipe on one of a battery of boilers should split would it be practicable to increase the speed of the feed pump to make up for the leakage?

S. B. P.

Probably not. The leakage from a short crack which could be prevented from extending to a dangerous length by suitable clamps, might be made up by an increase in pump speed. But in case of a serious rupture, the boiler should be cut from the line and the fire covered.

Steam Pressure in Boiler

In the issue of October 4, P. S. B. asks: Is the pressure equal in all parts of a steam boiler? A part of your reply is that the pressure of the steam is exerted equally in all directions on the water as well as on the shell.

I find in Roper's "Handbook of Land and Marine Engines" the following statement in the chapter relating to longitudinal and curvilinear strains: "The strain on a boiler subjected to internal pressure transversely is exactly double what it is longitudinally, or in other words, the strain on the longitudinal seams is double that on the curvilinear." I should like your explanation of this matter.

S. O. S.

The pressure of steam in a boiler is equal in all directions, but the stress on the curvilinear seams per inch of length is one-half that on the longitudinal ones.

The stress on each inch of the longitudinal seam is,

$$Radius \times pressure = stress,$$

while that on each inch of the curvilinear seam is,

$$\frac{Area\ of\ head \times pressure}{3.1416 \times diameter} = stress$$

In a boiler 60 inches in diameter carrying 100 pounds pressure per square inch, the stress on each inch of the longitudinal seam is

$$30 \times 100 = 3000\ pounds,$$

while on each inch of the curvilinear seam it is

$$\frac{2827.4 \times 100}{188.496} = 1500\ pounds$$

Area to be Stayed

What is the area to be stayed on a 60-inch boiler? Distance from tubes to shell 21 inches?

A. B. S.

The area of a segment of a boiler head to be braced is the area inclosed by lines drawn 3 inches from the shell and 2 inches from the tubes and is found approximately by the formula

$$Area = \frac{4H^2}{3}\sqrt{\frac{2R}{H} - 0.608}$$

in which,

H = Distance from tubes to shell less 5 inches;

R = Radius of boiler less 3 inches.

Substituting

$$\frac{4 \times (16)^2}{3}\sqrt{\frac{2 \times 27}{16} - 0.608} = 560.25$$

square inches of area to be stayed.

Trouble with Governor Safety Pin

I have charge of a Corliss engine upon which, as soon as the governor leaves the safety pin, it will drop and stop the engine if I do not hold it up.

This occurs only when starting up. Can you give me the cause of the trouble?

A. H. B.

You have slightly misstated the case. The governor does not leave the safety pin; that is, it does not rise. As the speed of the engine increases, the pressure of the governor rest on the safety pin is reduced, and as the pin is held in place by the friction between it and the governor rest, the force which moves the pin aside overcomes the reduced friction, and the pin is moved from under the rest before the governor has acquired the speed necessary to make it rise; therefore, it drops. The remedy is to make a shallow recess in the safety pin, out of which the governor rest must be lifted by rising from the effect of centrifugal force before the pin is moved aside.

Efficiency of Riveted Seams

What is the efficiency of joint? Thickness of plate ⅜ inch, 55,000 pounds tensile strength $\frac{13}{16}$ inch diameter of rivets, pitch 3½ inches, joint single riveted?

E. R. S.

The efficiency of a riveted seam is

$$Efficiency = \frac{p - d}{p}$$

in which

p = Pitch of rivets in inches;

d = Diameter of rivet in inches;

hence,

$$\frac{3.5 - 0.8125}{3.5} = 0.767$$

or 76.7 per cent., the efficiency of the joint, and its strength is

$$0.375 \times 55{,}000 \times 0.767 = 15{,}819.37$$

pounds per inch of length.

Safe Working Boiler Pressure

What is the safe working pressure of a boiler 60 inches diameter, 55,000 pounds tensile strength, ⅜ inch thickness of plate, efficiency of seam, 70 per cent.; factor of safety, 5?

W. B. P.

The working pressure on boiler shells and drums is determined by the formula

$$\frac{T.S. \times t \times \%}{R \times F.S.} = working\ pressure$$

in which

$T.S.$ = Tensile strength of plate in pounds per square inch;

t = Thickness of plate in inches;

$\%$ = Efficiency of seam;

R = Radius of shell;

$F.S.$ = Factor of safety.

Substituting

$$\frac{55{,}000 \times 0.375 \times 0.70}{30 \times 5} = 96.25$$

pounds per square inch working pressure.

Safe Flywheel Speed

I wish to run an engine at 200 revolutions per minute; what will be the diameter of the largest cast-iron flywheel I can use with safety?

S. F. S.

The rim speed of a cast-iron wheel should not exceed 6000 feet per minute. The diameter of a wheel for this speed may be found by dividing 1910 by the number of revolutions.

$$\frac{1910}{200} = 9.55\ feet$$

the safe diameter of a flywheel for 200 revolutions per minute.

Calculating Pipe Sizes

Is there any simple rule for calculating the size of a pipe which shall have an area equal to that of two other pipes of different sizes?

C. P. S.

Lay off on one side of a square a distance equal to the diameter of one of the pipes and on the adjacent side a distance equal to the diameter of the other. The diagonal distance across the corner of the square from one point to the other is the diameter of a pipe the area of which will equal that of both the others.

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November 1..... 33,000
November 8..... 33,000

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Power and The Engineer

Editorial

Unification of the Steam Turbine

The steam turbine seems to be growing toward a general type built by all the large makers. We found that practically all the European builders of the Parsons type, except Parsons himself, and several builders of the Rateau type are using a velocity stage, i.e., the Curtis system, for the initial expansion. In the Parsons turbine a large part of the barrel is made up of short blades of practically the same length to effect the same expansion which takes place in the first stage of a Curtis. By allowing this expansion to take place in a single set of nozzles and using a Curtis set of two rows of running and a single row of reversing buckets to absorb the velocity so generated, the steam is reduced in pressure from, say, one hundred and fifty to thirty pounds before it is introduced to the turbine casing, decreasing the pressure upon the balancing plates or dummies, diminishing the temperature to which the rotor drum and shell are subjected, and eliminating the long section of short blades, admittedly the least efficient of the turbine on account of their large windage in the dense medium of the high-pressure steam and of the large proportion of their clearance to the active surface. The doing away with this section also greatly diminishes the length of the turbine between bearings and gives a stiffer and more manageable shaft or drum, which allows of higher rotative speeds. In the Rateau type the large initial fall of pressure brings so much less pressure upon the stuffing box at the high-pressure end. Professor Rateau himself, however, does not approve of the use of the Curtis member here, preferring to make the same drop, if desirable, in a divergent nozzle before the steam comes into the casing and against the stuffing box, but absorbing as much of the consequent velocity as possible with a single row of blades, preserving the residual velocity for use upon the second wheel by the entrance curve of the second row of nozzles, and augmenting that velocity by expansion in that second row until the work performed by that row is substantially the same as that in the first. Even he, however, uses the velocity-stage construction where lightness and first cost are the main considerations. On the other hand, the British Thomson-Houston Company and the Allgemeine Elektricitäts Gesellschaft, the builders of the Curtis type in Great Britain and Germany respectively, have used the Curtis reversing blades and second row of runners only in the earlier stages of many of their turbines, using single runners in the lower pressure stages. In our own country the Westinghouse Machine Company uses the double-runner velocity stage for the initial expansion, not of all but of some of their turbines, while we understand that the General Electric Company, the builders of the Curtis type, have built some units in which the Parsons system is used for the final stages.

The systems appear therefore to be growing together by the adoption of the other builders of Curtis initial stages and the renouncing of the Curtis builders of the Curtis principle for the final stages.

Professor Beluzzo has proposed and Gadda Finzi & Co., of Milan, have built a unit in which a Curtis velocity stage is used for the initial expansion, Rateau pressure stages for the middle range and Parsons full admission reaction stages for the final expansion.

Hotel Power Costs

For the past several months the *Hotel World* has been carrying on a controversy among its readers as to whether it is policy for a hotel to operate its private plant or to buy current from a central station.

From the several score of letters on the subject it is evident that a great majority of managers have no definite information on the subject. Their decisions are based largely on precedent and frequently have no better foundation than "We operate a private plant because we found it in the basement when we bought the hotel," or "We buy central-station current because we always have." Apparently there has been little investigation of the subject from a commercial standpoint and there is a surprisingly large amount of sentiment attached to the decisions where sentiment should not exist. Engineering matters above all others should be reduced to a commercial basis. It is easy to do this if the proper arrangements are made, but we must acknowledge that it is not always done even in plants of much greater importance than the local installation in a hotel building.

It is pointed out that the problem of a power plant for a hotel is much more involved and complicated than for the ordinary office building. There is the

question of live steam for steam tables in the restaurants; power for a great variety of purposes as well as lighting and heating; refrigeration in all its modern applications, ice, water and other services which do not enter into the ordinary building requiring power. However, in just so much as the problem is complicated it should receive all the more the careful attention of the manager and engineer, and there is no excuse for not knowing in dollars and cents just what the plant is doing and why it is the best solution of the problem under the circumstances.

Out of the mass of correspondents, however, one communication looms up with exceptional brilliancy. This is from J. H. Morrow, chief engineer of the Great Northern hotel, Chicago, who gives definite figures as to operating costs both before and after installing an isolated plant.

For the year 1905, to quote from the report, the expense of the plant while buying current from a central station was:

Light and power	$38,563.01
Fuel	9,502.65
Water	1,943.40
Engineers and machinists supplies	6,291.34
Salaries	8,540.99
Total	$64,841.39

In 1906 with a private plant in operation the figures reached were as follows:

Fuel	$15,512.43
Water	984.90
Engineers and machinists' supplies	4,140.51
Salaries	12,647.87
Total	$33,285.71

These figures show a net saving of $31,555.68 or a percentage of 48.5 in favor of the isolated plant.

In the report it is stated that in the year 1906 the private plant generated 312,894 kilowatt-hours in excess of the power used in 1905, making the performance in reality still more favorable to the isolated plant. Figures for the ensuing year up to 1910 show the same character of performance.

It is stated in connection with this plant that the average cost per kilowatt-hour for the last four years has been 2.49 cents, the price of coal during this period ranging from $2.45 to $2.55 per ton. Adding the capital charges to the plant still leaves the total cost per kilowatt-hour under three cents.

This operating cost it must be understood includes every expense attached to the power plant such as fuel, salaries, electric and machinery supplies, water, lamp renewals, pumping water, elevator service, steam for all auxiliaries, including kitchen service, refrigerating plant, steam vacuum-cleaning system, air compressors and for heating the building.

It is gratifying to note that in this instance the matter of power costs has been gone into in such a thorough manner and the incident serves as a shining example to other engineers placed in similar positions or having charge of any kind of a power plant where this subject has not been given the attention it deserves.

Central Station vs. Isolated Plant

As an illustration of one of the ways in which a thing should not be done, the meeting held under the auspices of the Blue Room Engineering Society at the Engineering Societies' building, New York, on the evening of October 9, was complete.

The meeting was called for the purpose of discussing the encroachment of the central station upon the field of the isolated plant, and, if possible, of showing to the public in general, and to the operating engineer in particular, that in the great majority of cases the owners of office, manufacturing and apartment buildings can make the electric current which they use for light and power cheaper than it can be bought. Had the meeting been controlled by the most skilful manipulator in the service of the Public Service Corporation, the tendency to discourage further discussion could not have been increased.

In the language of the player folk—"It was a frost." Announced for eight o'clock, the meeting was called to order at ten minutes of nine, when the chairman introduced Charles G. Armstrong, who for more than an hour read extracts from his report to the Bureau of Municipal Research, and explained the meaning of the charts which had been distributed to the audience. The common, or garden variety of operating engineer in whose interest the meeting was supposedly held, is not a chart-and-formula-loving animal, and while he is in thorough sympathy with the argument and appreciative of the facts presented, he commenced to chafe under an hour's application of this treatment.

Charles M. Ripley, whose association with the Engineering Supervision Company as its vice-president, would have rendered it difficult for him to gain the sympathy of his audience under the best of circumstances, for the engineer dreads supervision as much as he does the central station, followed and took up another hour in the wearisome presentation, detail by detail and exhibit by exhibit, of a long argument, the essentials of which were available in print and which needed only an abstract presentation. With an utter lack of appreciation of the temper of his audience, and a woeful display of tact, he interspersed his remarks with such crass advertisements of the Supervision Company as to evoke audible expressions of disapproval, and in some cases outspoken protests.

It is unquestionable that in the majority of buildings in New York City, heat, light and power can be furnished on the premises for less than it can be bought from any outside source, however well equipped, but it is not the engineer who needs educating along this line, it is the owner and the architect who must be convinced.

Owners are much like other human beings, and are like others influenced by their economic interests. Not one will use central-station service when he can get the same service for less money elsewhere if he knows it. It is up to the engineer to show where this service can be got for the least money, and unless his is one of the exceptional plants in which this is not the case, show that he can do it in his own plant better than it can be done elsewhere.

Striking Heat Conversions

It is a striking demonstration of the conversion of heat into mechanical energy to stand beside a steam turbine and place one's hand upon the cool exhaust pipe on one side while a few inches upstream on the boiler side of the nozzles the pipe is so hot that it cannot be touched. In flowing through these few inches the steam has dropped upward of two hundred and fifty degrees in temperature and converted from ten to fifteen or more per cent. of its molecular kinetic energy into useful work.

A more striking conversion, however, takes place in the boiler where in the gas surrounding the heating surface there is a temperature of some two thousand degrees while the exterior surface of the iron has a temperature not many degrees above that of the water within.

In a film of a thousandth of an inch in thickness the velocity of the gas molecules is reduced to that corresponding (in connection with their mass) to the lower temperature, and the velocity of a mass of iron and of water molecules is accelerated until the energy required for such acceleration equals that given up by the gas molecules in their retardation.

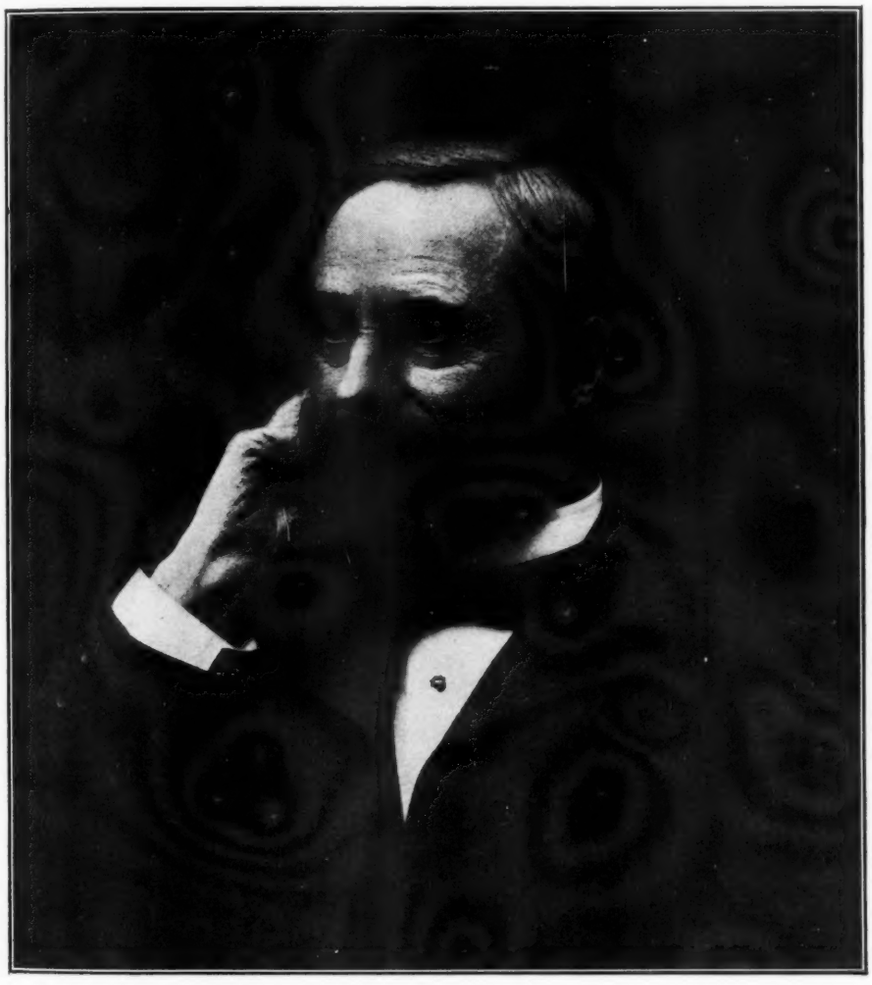
The Tall Chimney

In discussing the report of the Committee on Power Generation, of the American Street and Interurban Railway Engineering Association, at Atlantic City recently, C. O. Mailloux expressed himself very forcibly as of the opinion that the day of the tall chimney is passing and that draft will be furnished by mechanical means. Upon the editorial page of POWER for May, 1890, the opinion was expressed that "the time will come when these tall chimneys will serve no more useful purpose than that of monuments to the folly of their builders."

The fuel used in manufacturing costs too much to allow twenty-five per cent. of it to escape up the stack "to warm up outdoors," and the mechanical engineer whom we were quoting opined even at that time that "future development will be in the direction of regaining from the flue gases every possible unit of heat, and the maintenance of the draft, not by the maintenance of a tall column of hot gas, but directly by mechanical means."

Dr. Stillman W. Robinson Dead

On Monday morning, October 31, Prof. Stillman Williams Robinson, inventor and engineering expert, died at his home in Columbus, O. He was born on a farm in Reading, Vt., March 6, 1838. As a boy he took great interest in matters mechanical, but what with farm work and the poor educational facilities in a country place he had slight opportunity to follow his inclination. He studied and invented in a small way, however, and when he became fifteen years old his opportunity to give up farming presented itself, and his sixteenth year was spent in operating a sawmill and erecting a furniture factory and a gristmill. At the end of that year he entered upon a four years' apprenticeship in the machine shop of F. B. Gilman, Springfield, Vt. He studied during this period, meeting his school expenses by occasional jobs at the trade he was learning.



DR. STILLMAN W. ROBINSON

In 1860, with eight dollars and a kit of stencil tools, young Robinson started for Ann Arbor, Mich., to attend the State University there. He defrayed the expenses of the trip by cutting stencils from town to town, and reached Ann Arbor with fifty dollars in his pocket. During his college career he supported himself by making stethoscopes and graduating thermometer scales. He was graduated in 1863, with the degree of civil engineer. On leaving the university he became assistant engineer for the United States Lake Survey, acting in that capacity from 1863 until 1866. In the latter year he became instructor in mechanical and civil engineering at the University of Michigan, and from 1867 to 1870 he was instructor in mining engineering and geodesy at the same institution. In 1870 he was appointed professor of mechanical engineering and physics at the University of Illinois, the first school of mechanical engineering in the country, where he remained until 1878, in the last-named year being dean of the college of engineering at this university. From 1878 to 1891 he was professor of mechanical engineering and physics in the Ohio State University, and from 1891 to 1895 professor of mechanical engineering.

In 1895 Professor Robinson resigned to become inventor and consulting mechanical engineer for the Wire Grip and Fastener Company and the McKay Shoe Machinery Company. His first invention was a thermometer-graduating machine, made while in college. Various other inventions, notably a number for shoe machines, yielded him about forty patents. He was adverse to the use of cams in machine elements and was peculiarly successful in designing link and lever motions. Many of his machines are in use today.

In addition to his other work he was State inspector of railroads and bridges from 1880 to 1884, consulting engineer to the Santa Fé road from 1887 to 1890, and consulting engineer of the Lick telescope mountings in 1887.

In 1896 the degree of doctor of science was conferred upon Professor Robinson by the Ohio State University, and in 1899 he was elected to the position of professor emeritus in mechanical engineering.

Professor Robinson was a member of the American Society of Mechanical Engineers, American Society of Civil Engineers, and of the Naval Architects and Marine Engineers; a fellow of the American Association for the Advancement of Science, and of the Society for the Promotion of Engineering Education. He was author of a number of technical works, among them "Principles of Mechanism," a college textbook, besides being a contributor of engineering literature to the technical press.

At the time of his death he was president of the S. W. Robinson and Son Company of his home city.

In personality he was kind and genial, always ready with helpful suggestions, particularly for the young men around him. His stimulating influence will be recalled with gratitude by many engineers who came in contact with him either as students or in business.

The story of his rise from the obscurity, poverty and hardship incidental to the life of the New England farm-born lad to the position of a master of engineering through his own unaided personal efforts is peculiarly American in its character and should serve as an incentive, if not an inspiration, to all who read of his life's work. Of him it can be most fittingly said:

"Lives of great men all remind us
We can make our lives sublime
And departing leave behind us
Footprints on the sands of time."

New Power House Equipment

Elliott Reducing Valves

The accompanying illustrations show the design of types *A* and *B* reducing valves made by the Elliott Company, 6907 Susquehanna street, Pittsburg, Penn. The operation of either valve is as follows:

The steam or other fluids pass through either valve, Figs. 1 and 2, in the direction indicated by the arrows, the high-pressure steam entering at *H*, and at reduced pressure is discharged at *S*. The main steam valve *A* is connected to the spindle *B* with the two case caps *C* sur-

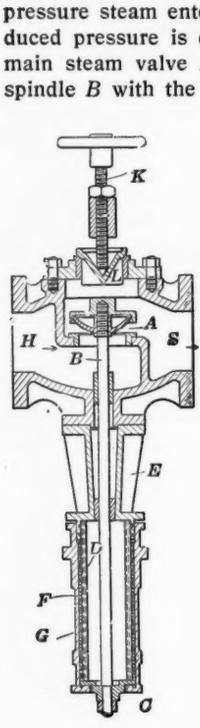


FIG. 1

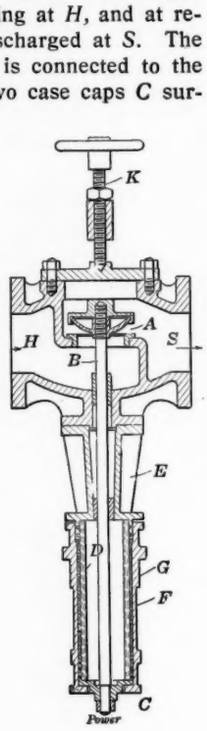


FIG. 2.

rounding the india-rubber tube *D*, which rests in the bottom, as shown, and is also accommodated in a similar space in the bottom of the extension or distance piece *E*. This india-rubber tube is reinforced on the outside by a number of neatly fitting rings *F*, as shown in Fig. 3. The object of these rings is to permit the tube to stretch only in the direction of its length, keeping it constant in diameter (see second illustration, Fig. 3), and at the same time eliminating friction by the tube coming in contact with the tube case *G*, Figs. 1 and 2. By making the tube *D* the exact diameter of the seat of the valve *A*, the valve is put in equilibrium to any pressure acting in the direction of the arrows.

The tube-case cap *C* rests in the tube case *G*, which is provided at the proper end with wings *H*, shown in Fig. 4, to receive the upward pull of the springs *I* which are put in tension by the yoke *J* and adjusting screw *K*. The pull on the spring disturbs the equilibrium established

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

and forces the valve *A* upward and the amount of tension determines the reduction in steam pressure.

If the area of the valve *A* is 1 square inch and the upward pull on the spring is 10 pounds as soon as the steam pressure on top of the valve equals 10 pounds, the weight will overcome the spring and the india-rubber tube will commence to stretch and allow the valve *A* to close, thus preventing further passage of steam. The india-rubber tube *D* with its surrounding wings is in effect an elastic cylinder, the bottom end of the tube-case cap *C* acting as a piston. The move-

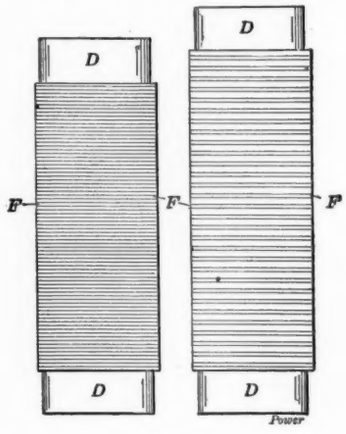


FIG. 3

ment is frictionless, as the piston does not move in the cylinder, but the body of this cylinder or rubber tube is itself elongated. The movement of the tube being divided among the rings *F*, the tube even when extended is so perfectly supported as not to be distressed in the slightest, it is claimed. The tension of the springs, therefore, controls the reduced pressure which can be varied by adjusting the screw *K*.

The pressure on the adjusting screw *K* is received by the valve *L*. This valve *L* is made with about 20 per cent. less diameter than the valve *A*, and will lift if the pressure above the valve *A* exceeds by 20 per cent. the pressure for which the reducing valve is set. The valve *L*, therefore, acts as a relief valve against overpressure should the vale *A* fail to shut off the steam.

The rubber tube is protected from injury from steam by being filled with condensed steam. Under ordinary working conditions the tube will last for years, it is said. In case the tube should prove defective a new one can be inserted by simply decreasing the tension on the springs and removing the yoke. The tube can then be taken from the tube casing.

Type *B* reducing valve, Fig. 2, is in every way similar to type *A* with the ex-

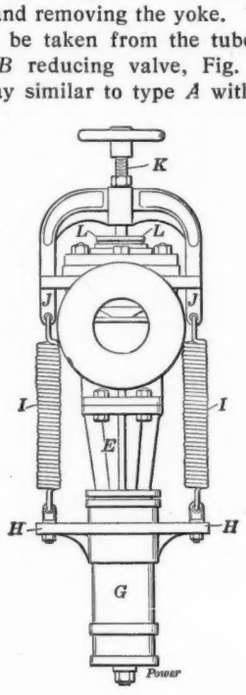


FIG. 4

ception that no safety valve is provided for the escape of the excess pressure.

These valves are made in sizes ranging from ½ inch up to and including 4 inches, with either flange or screw connections, and are capable of working under initial pressure of 250 pounds per square inch. The valves up to and including 1½ inches are made of gun bronze; above that of cast iron with bronze wearing parts.

The Manufacture of Humphrey Pumps

Brown Boverie & Co., of Baden, Switzerland, have just completed an arrangement with the Pump and Power Company, Limited, whereby they become the licensees of that company for the manufacture of the Humphrey pump and compressor. The Humphrey pump, it will be recalled, is that in which a charge of gas is exploded in contact with the water to be pumped, and the functions of compression, etc., are effected by the surge of the water column. The same

arrangement is used as an air compressor by allowing the other end of the reciprocating water column to draw air into a separate chamber, situated at the distant end of the discharge pipe and then compress it and deliver it under pressure on the next explosive charge.

Siemens Schuckertwerke are making at Berlin, one of these pumps which will deliver a million gallons of water per hour to a hight of 200 feet. One of the new experimental pumps to be erected at Dudley Port, Eng., will have a suction lift of 18 feet and a direct line of 160 feet. The Pump and Power Company, Limited, has in hand for the Indian government some small pumps with suction lift, and among the new types, which they are developing, is a well pump in which practically the whole lift is suction.

Friction Test upon Keystone Grease

The figures shown in the accompanying table represent the average results of a test upon a sample of Keystone grease as recently conducted with a Cornell oil-testing machine at the testing laboratory of the William Cramp & Sons Ship and Engine Building Company.

The test consisted of runs at various pressures, each run being of one hour's duration with readings taken at five-minute intervals and fifteen minutes allowed between runs for the temperature of the journal to become constant. The journal was of a standard shape, having an area of 14 square inches and filled with Cramp's government babbitt and grooved diagonally. The grease was fed from a hand-pressure grease cup of about three ounces capacity.

FRICTION TEST.

Run.	Room Temperature, Degrees Fahrenheit	Journal Temperature, Degrees Fahrenheit	Increase in Temperature of Journal over Room.	Friction in Pounds.	Pressure in Pounds.	Average Coefficient of Friction.
No. 1	78.6	157.7	79.1	11.9	2000	0.00595
No. 2	79.2	170.3	91.1	16.4	3000	0.00547
No. 3	80.3	176.1	95.7	15.3	4000	0.00333
No. 4	83.0	185.2	102.2	16.3	5000	0.00326

The speed of the shaft was 450 revolutions per minute and during each entire run the grease maintained its original consistency.

NEW PUBLICATIONS

ELECTRIC WIRING. By Joseph G. Branch. Published by the Branch Publishing Company, Chicago, 1910. Cloth; 272 pages, 5x7½ inches; 93 illustrations; numerous tables. Price, $2 net.

The subject is treated in the form of a catchism; beginning with the elementary consideration of a simple current flowing through a single conductor, and leading up to the more involved aspects of the subject. Problems are frequently employed to illustrate points under consideration and application is made of the various formulas commonly used in making wiring calculations. For the greater part low-potential wiring is dealt with, such as that employed for the lighting and power circuits of buildings or small towns; and only casual reference is made to high-potential wiring.

In point of criticism, it may be said that many of the definitions are extremely loose and would be apt to mislead one not already familiar with electricity. Also, the text contains numerous statements which are contradictory and do not conform to uptodate practice.

SOCIETY NOTES

During the month of November at the Modern Science Club of Brooklyn, the following lectures will be given: "The Panama Canal," by N. B. Payne, November 15; "Injectors," by Frank O'Leary, November 22; "A Modern Hydraulic Installation," by George A. Orrok, November 29.

At the regular annual meeting of the Association of Edison Purchasing Agents the present officers were elected to hold office from 1910 until 1911: H. C. Lucas, president; W. H. Francis, first vice-president; J. W. Brennan, second vice-president; A. W. Banks, Jr., treasurer; J. J. Miley, stock controller.

The monthly meeting of the Engineers of Boston will be in charge of the American Society of Mechanical Engineers, the Boston section of the American Institute of Electrical Engineers and the Boston Society of Civil Engineers coöperating. The meeting will be devoted to a topical discussion of a paper written by D. T. Randall, engineer in charge of fuel testing with A. D. Little Company, on the subject of "Smoke Abatement."

A paper on the "Rotary Kiln," by Ellis Soper, of Detroit, will be presented at the New York meeting of the American Society of Mechanical Engineers, 29 West Thirty-ninth street, on November 9. Following this paper, Charles Whiting Baker, editor of *Engineering News*, will give an illustrated lecture on the Panama canal. The date of this meeting has been changed from the customary one because the meeting would have come this month on election day.

The sixth annual entertainment and reception of the Universal Craftsmen Council of Engineers, of the Metropolitan district, including the boroughs of Manhattan, Brooklyn, Queens, Bronx and Richmond and the cities of Newark, Jersey City and Paterson, N. J., was held at Lexington Opera House, New York City, on Saturday evening, October 29. There was a large attendance, and the usual merry time was had. An enjoyable vaudeville entertainment was followed by a long dancing program.

Secretary Calvin W. Rice, of the American Society of Mechanical Engineers has just returned from a trip to San Francisco, where he attended an organization of the local members for the purpose of holding meetings in that city. A. M. Hunt was elected chairman and T. W. Ransom secretary of the local organization, with Prof. W. F. Durand, E. C. Jones and Thomas Morrin as an executive committee. A resolution was adopted inviting the American Society of Mechanical Engineers to hold its semi-annual meeting in San Francisco during the Panama-Pacific Exposition of 1915, and the suggestion was made to the council of the society that an international meeting of mechanical-engineering societies be held conjointly with that of the American society. It has not yet been decided, however, that the international exposition of 1915 will be held in San Francisco, although that city is making very earnest efforts for it, including the passage of a constitutional amendment, enabling the State to raise $5,000,000 by taxation for this purpose.

PERSONAL

Charles Guest Norris, who has been in this country for a number of months, disposing of American interests in English inventions, entertained a party at the Engineers' Club on Wednesday evening, October 26, on the eve of his return to England.

S. B. Whinery, manufacturers' agent, 95 Liberty street, New York City, announces that he has recently made arrangements with the American Well Works, of Aurora, Ill., to represent them in the New York district in the sale of centrifugal and turbine pumping machinery.

R. T. Hudson, formerly shift engineer at Norwich Electricity Company, Norfolk, Eng., and for the past five years shift engineer at the municipal power station of Johannesburg, S. A., has lately been appointed shift engineer at the Randfontein Estates Gold Mining Company's central power station, Transvaal, S. A., which has lately been equipped with Parsons turbo-alternators and has an aggregate capacity of 20,000 kilowatts, the largest units being two 6000-kilowatt sets.

Cost of Power

On Saturday evening, October 29, a public mass meeting was held in the auditorium of the United Engineering Societies' building, New York City, under the auspices of the Blue Room Engineering Society, to listen to addresses by Charles G. Armstrong, mechanical engineer for New York City, and C. M. Ripley.

Mr. Armstrong's address was based on an investigation which he has been conducting during the last three months for the Bureau of Municipal Research for the purpose of ascertaining the relative cost of light, heat and power when the electric current is furnished by the central station and when it is furnished by the isolated plant. An abstract of Mr. Armstrong's address will be offered in an early issue.

Mr. Ripley's paper on "The Cost of Heat, Light and Power," was presented in the issue of POWER for October 25.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE. Theodore H. Haberkorn, Fort Wayne, Ind. 973,651.

GAS ENGINE. Charles M. Leech, Lima, Ohio. 973,792.

INTERNAL COMBUSTION MOTOR. Donald J. McKinnon, Toronto, Ontario, Canada, assignor of two-thirds to K. F. McKinnon and Ernest W. Oliver, Toronto, Canada, and William A. McKinnon, Birmingham, England. 973,800.

WATER MOTOR. Norman A. Stamm, Glendora, Cal. 973,823.

ROTARY GAS ENGINE. Herbert I. Wilber, Dorchester, Mass. 973,833.

TIDE-WATER MOTOR. Martin Logan, New York, N. Y. 973,869.

TURBINE. Gustav Engisch, Berlin, Germany. 973,929.

BOILERS, FURNACES AND GAS PRODUCERS

OIL VAPORIZER AND BURNER. Oral A. Makinson, Pratt, Kan. 973,526.

WATER-TUBE BOILER. Jiro Miyabara, Tokyo, Japan. 973,541.

DEVICE FOR INCREASING HEAT. Gamaliel C. St. John and Frederic A. D. Rankins, New York, N. Y. 973,581.

FURNACE. Gilbert R. Haigh and John R. Fortune, Detroit, Mich., assignors to Murphy Iron Works, Detroit, Mich., a Corporation of Michigan. 973,652.

STEAM BOILER. Carl Marinius Olsen, Birkerød, near Sandefjord, Norway. 973,692.

SUPERHEATER BOILER. James P. Sneddon, Barberton, Ohio, assignor to the Babcock & Wilcox Company, New York, N. Y., a Corporation of New Jersey. 973,717.

VERTICAL BOILER. Tozaburo Suzuki, Sunamura, Japan. 973,720.

GAS-GENERATING OIL BURNER. Thomas Muehleisen, San Diego, Cal. 973,804.

FURNACE. Luther L. Knox, Avalon, Penn., assignor to Keystone Furnace Construction Company, Pittsburg, Penn., a Corporation of Pennsylvania. 973,943.

SMOKE-CONSUMING APPARATUS FOR LOCOMOTIVES. John Loftus, Albany, N. Y., assignor of one-half to Anthony Hughes, Philadelphia, Penn. 973,947.

SMOKE-PREVENTING AND FUEL-ECONOMIZING APPLIANCE. William John Storey and Gilbert Storey, Albert Park, near Melbourne, Victoria, Australia. 973,991.

LIQUID FUEL BURNER. Charles A. Swartz and Sam Finley, Frederick, Okla. 973,995.

POWER PLANT AUXILIARIES AND APPLIANCES

DETACHABLE BOILER FLUE. Ole C. Borgen, Sharon, N. D. 973,471.

HYDRAULIC PUMP. Robert Cairns, Toronto, Ontario, Canada. 973,473.

HOSE CLAMP. Milton P. McLaughlin, Wakefield, Mass., assignor of forty-five one-hundredths to H. C. McCarty, Williamsport, Penn. 973,532.

VALVE. William L. Osborne, Chicago, Ill. 973,551.

CARBURETER. Joseph G. Williams, Buffalo, N. Y. 973,602.

TUBE AND HEADER CONNECTION. James C. Alexander, Roseburg, Ore. 973,610.

STEAM SUPERHEATER. Henry W. Jacobs, Topeka, Kan. 973,662.

BLOWOFF VALVE. Henry W. Jacobs, Topeka, Kan. 973,663.

PUMP, ETC. Fitz W. Machlet, Elizabeth, N. J. 973,679.

GOVERNING MECHANISM FOR MIXED PRESSURE TURBINES. Frederick Samuelson, Rugby, England, assignor to General Electric Company, a Corporation of New York. 973,705.

TUBE CLEANER. Thomas Andrews, Rockaway, N. J. 973,740.

CARBURETER. William C. Carter, St. Louis, Mo. 973,755.

LIQUID FUEL STRAINER. William C. Carter, St. Louis, Mo. 973,756.

WATER HEATER AND PURIFIER. William Asa Gibson, Peoria, Ill. 973,773.

CENTRIFUGAL FORCE PUMP. Thomas Russell Hayton, Mount Vernon, Wash. 973,782.

CARBURETER. Percival Charles Cannon, London, England, assignor to Montague Stanley Napier and Selwyn Francis Edge, London, England. 973,855.

SPARK PLUG FOR EXPLOSIVE ENGINES. Martin T. Minogue, Philadelphia, Penn., assignor, by direct and mesne assignments, to The Superior Motor Specialty Company, New York, N. Y., a Corporation of New York. 973,873.

CARBURETER. Burt Neulon Pierce, Indianapolis, Ind. 973,877.

CARBURETER. William F. Rothe, East St. Louis, Ill., assignor to Wm. F. Rothe & Company, East St. Louis, Ill., a Corporation of Illinois. 973,882.

CARBURETER. William Ernest Haines, Cheltenham, England. 973,937.

PACKING FOR FLUID ENGINES. William F. Purcell, Brooklyn, N. Y. 973,972.

SCALE PREVENTER FOR BOILERS. John W. Stephens, Staunton, Va. 973,989.

FIRING INDICATOR AND RECORDER. Asa White Kenney Bilings, Habana, Cuba. 974,017.

ELECTRICAL INVENTIONS AND APPLICATIONS

INCANDESCENT ELECTRIC LAMP. Thomas E. Robertson, Catford, England, assignor to General Electric Company, a Corporation of New York. 973,703.

ELECTRIC MOTOR. Edwin S. Pillsbury, St. Louis, Mo., assignor to Century Electric Company, a Corporation of Missouri. 973,878.

CIRCUIT CONTROLLER. Winfield A. Atwood, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 973,613.

TERMINAL FOR ELECTRICAL APPARATUS. William B. Potter, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 973,560.

SELECTOR SWITCH. Edward B. Craft, Chicago, Ill., assignor to Western Electric Company, Chicago, Ill., a Corporation of Illinois. 973,762.

CIRCUIT-CLOSING DEVICE. Leslie R. Saunders, Los Angeles, Cal. 973,818.

ELECTRICAL OSCILLATOR. Frederick K. Vreeland, Montclair, N. J., assignor to Wireless Telegraph Exploitation Company, New York N. Y., a Corporation of New York. 973,826.

CURRENT CONTROLLER. Frank L. Sessions, Columbus, Ohio, assignor, by mesne assignments, to the Jeffrey Manufacturing Company, a Corporation of Ohio. 973,885.

ELECTRICAL CIRCUIT BREAKER. Harve R. Stuart, Wilkinsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 973,889.

ELECTRIC BRAKE. Robert R. Dunlop, Columbus, Ohio, assignor, by mesne assignments, to the Jeffrey Manufacturing Company, a Corporation of Ohio. 973,926.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.